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Research on Control Technology for Ice Fog From Mobile Sources

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RESEARCH ON CONTROL TECHNOLOGY FOR ICE FOG FROM MOBILE SOURCES

by

Harold J. Coutts
Ronald K. Turner
Arctic Environmental Research Station
Corvallis Environmental Research Laboratory
College, Alaska 99701

CORVALLIS ENVIRONMENTAL RESEARCH LABORATORY
OFFICE OF RESEARCH AND DEVELOPMENT
U. S. ENVIRONMENTAL PROTECTION AGENCY
CORVALLIS, OREGON 97330

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FOREWORD

Effective regulatory and enforcement actions by the Environmental Protection Agency would be virtually impossible without sound scientific data on pollutants and their impact on environmental stability and human health. Responsibility for building this data base has been assigned to EPA's Office of Research and Development and its 15 major field installations, one of which is the Corvallis Environmental Research Laboratory (CERL).

The primary mission of the Corvallis Laboratory is research on the effects of environmental pollutants on terrestrial, freshwater, and marine ecosystems; the behavior, effects and control of pollutants in lake systems; and the development of predictive models on the movement of pollutants in the biosphere. CERL's Arctic Environmental Research Station conducts research on the effects of pollutants on Arctic and sub-Arctic freshwater, marine water and terrestrial system; and develops and demonstrates pollution control technology for cold-climate regions.

This report describes a two winter investigation of technology for controlling ice fog emmissions from mobile sources.

A. F. Bartsch
Director, CERL

ABSTRACT

Automotive generated ice fog is a form of air pollution that results when exhaust water vapor freezes into minute particles which form a dense fog. This study on control techniques was conducted by the U.S. Environmental Protection Agency at its Arctic Environmental Research Station near Fairbanks, Alaska.

The major control technique evaluated was the cooling of exhaust gases to well below the dew point, thus condensing water vapor into a liquid stream before final discharge.

During the winter of 1974-75 nine exhaust gas cooler-condensers were installed on local vehicles and their water vapor removal performances were evaluated. Based on these data three cooler-condensers were fabricated, installed, and more intensely evaluated during the winter of 1975-76. The sizing criteria developed the first winter were found inadequate because ice film formation decreased heat transfer efficiency. Cooler-condensers must be designed to avoid or to accommodate condensate freezing.

An ice fog mass emission reduction up to 80 percent was attained with cooler-condensers on motor vehicles. However, the increase in visibility over roads was not quite proportional because of the many other ice fog sources. The overall impact of automotive ice fog control would be a visibility increase of at least 70 percent in areas where motor vehicles create 50 percent or more of the ice fog.

Control of automobile-generated ice fog would also mean cleaner air, but perhaps more ice on the road. Cleaner air would result because sulfur oxides and lead compounds would be absorbed in the condensate. This condensate, if allowed to drip from the cooler-condensers, would freeze onto the road and require a more intense snow removal effort.

This study has shown that cooler-consensers are effective ice fog control devices for mobile sources. The next step is to further evaluate and demonstrate the devices on fleet vehicles used in the dense ice fog areas.

This report covers a period of work from July 1974 to May 1976.

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SECTION 1

SUMMARY AND CONCLUSIONS

Ice fog is a form of air pollution caused by water vapor released into air too cold to retain the water in the vapor phase. Thus the vapor condenses into ice particles. In cold climates where this condition is common during the dark winter months, the reduced visibility is a major citizen complaint.

There are many strategies available to reduce water vapor emissions from all sources. The two main strategies for reducing automotive-generated ice fog are: (1) reducing combustion engine vehicle use or (2) reducing combustion engine water vapor output.

Successful implementation of the first strategy would require an effective mass transit network and/or use of electric powered vehicles. Such traffic modifications were not included in this study. The second strategy would require some means of dehydrating vehicle exhaust. This study has shown that cooler-condensers can eliminate up to 80 percent of the ice fog caused by vehicular emissions.

A research effort encompassing two winters was directed toward finding and evaluating methods to reduce automobile-generated ice fog. Four methods are discussed: (1) allow ice fog particles to form, then capture them with particulate traps; (2) capture the water vapor with a dessicant; (3) remove the water vapor as a liquid condensate by use of cooler-condensers; and (4) warm the ambient air with an exhaust dilutor-air heater thus allowing it to accept the dispersed exhaust water vapor without immediately forming ice fog. The first two methods would require such cumbersome equipment that they were considered impractical. Only methods 3 and 4 were evaluated.

Cooler-condensers, devices for removing water vapor from gases, were evaluated for their ice fog control possibilities. These cooler-condensers are heat exchangers in which the exhaust gas is cooled by cold ambient air or coolant from the automobile's radiator.

The first winter's effort was spent adapting, modifying and attaching existing heat exchangers to vehicle exhaust systems. With the resulting field performance data, the overall heat transfer coefficients were calculated. Also, independently, overall heat transfer coefficients were estimated from engineering data books. These latter coefficients resulted in required heat transfer surface areas much larger than those indicated by the first winter's field data. The second winter's experience indicated that the actual exchanger surface area required is closer to that calculated from coefficients estimated from the engineering data book. Therefore, it was found that for effective ice fog removal from a standard size gasoline-fueled light duty vehicle with a 250 cubic inch displacement engine a cooler-condenser would need 1.9 square meters (21 square feet) of heat transfer surface area.

It was found that ice film formation limited the performance of unprotected bare tube, air cooled, cooler-condensers with 1 cm (1/2 in.) tubes. Because of ice plugging problems, smaller tube sizes are not recommended. The cooler-condenser should be designed for efficient condensate drainage. Weep holes (at low points in the cooler-condenser system) were required to drain condensed water pockets which would otherwise freeze and block the outlet manifold.

Using a baffled automotive radiator as a cooler-condenser with antifreeze coolant might reduce the icing problem and make the lowest cost cooler-condenser. Because of increased engine vulnerability and unknown (first winter) radiator performance, they were not as thoroughly researched as the air cooled cooler-condensers.

Location of weep holes and the final exhaust outlet is critical, because air currents around the vehicle may cause exhaust gases to enter the passenger compartment and raise carbon monoxide levels.

With the cooler-condensers only, much of the resultant condensed water formed a fine mist; therefore, use of coalescers was necessary to transform the mist into liquid water.

One cooler-condenser was installed on a diesel sedan, but it did not perform as well as those on gasoline engines. Also, its coalescer became plugged with soot. Another type of ice fog control device evaluated was a perforated spiral wound flexible metal exhaust hose coiled under a vehicle and attached to its exhaust pipe. Its function was to serve as an exhaust dilutor-ambient air heater. It significantly reduced visible ice fog emission but raised the passenger compartment carbon monoxide levels. However, it was not as effective in reducing ice fog as were the cooler-condensers.

To quantify the actual on-the-road performance of a cooler-condenser, an equipped vehicle was driven through urban Fairbanks under normal winter driving conditions. The overall water vapor condensed (ice fog removed) was 81 percent at an ambient temperature of -16°C (4°F).

The increased exhaust system back pressure due to cooler-condensers was found to have an insignificant effect on fuel economy.

The authors estimate that mobile sources are responsible for about fifty percent of the Fairbanks ground level ice fog at temperatures below about -40°C, if cooling pond emissions are excluded. Ice fog control to yield 80 percent reduction on all vehicles in the Fairbanks area would therefore reduce ground level ice fog emissions by about 40 percent. Because of increased stationary source emission at lower ambient temperatures the overall ice fog reduction would be less, but the ice fog would not be as dense over roadways. The 40 percent reduction in ice fog emissions would theoretically yield a 67 percent increase in visibility.

The condensation process would also tend to remove toxic exhaust products such as sulfur oxides and lead compounds. The sulfur oxide removals varied from 1 to 20 percent. The lead removals varied from 6 to 49 percent of that in the gasoline. At 85 percent water vapor removal the condensate to be disposed of amounts to 0.8 gallon per gallon of gasoline burned.

If not captured by the automobile, the condensate ends up as ice on the road. This additional ice amounts to about 20 percent of the normal snowfall in Fairbanks, Alaska for the four coldest winter months. It would probably require more effort by the road maintenance crews to keep it from accumulating at some intersections.

This limited research study has shown that cooler-condensers can be effective in limiting ice fog from automobiles and trucks. However, the problem of controlling condensate freezing in the cooler-condensers was not thoroughly investigated. More experience is needed before possible regulatory action can be considered. A further evaluation and demonstration of the devices on fleet vehicles used in the dense ice fog areas should be carried out.

SECTION 2

RECOMMENDATIONS

Ice fog is a form of air pollution that becomes a problem when temperatures drop below -18°C (0°F). It is estimated that motor vehicle exhaust contributes nearly half the ground level ice fog.

Exhaust gas dehydration by use of water-condensers can effectively eliminate 75 percent of the automotive generated ice fog. If possible regulatory action is to be considered, then the next step would be to evaluate long term cooler-condenser operation and maintenance problems. This could best be accomplished by using the information supplied in this report to design and attach prototypes to 20 or more fleet vehicles that are routinely operated in dense ice fog areas. That evaluation could be used to determine which coolant works best and the degree of temperature controls needed to prevent cooler-condenser freeze ups. The result could also be used to derive a cost-benefit ratio for automotive ice fog control.

Because of the possibility of carbon monoxide poisoning, undercarriage exhaust discharge is not recommended for passenger vehicles.

Coalescers were necessary to eliminate mist emissions and are recommended for each vehicle equipped with a cooler-condenser.

SECTION 3

INTRODUCTION

BACKGROUND

Alaska is the largest, most sparsely populated, least industrialized state in the nation. Yet its major cities, Fairbanks and Anchorage, have winter time air pollution levels which rival those of New York and Los Angeles. The air quality of these Alaskan cities is degraded mainly by three types of pollutants: ice fog, other particulates, and carbon monoxide (1). The toxic health effects, if any, of ice fog have not been documented. Ice fog is most severe in Fairbanks but is increasing in Anchorage. This study deals with controls for ice fog from mobile sources. Ice fog air pollution is unique to regions with extremely cold climates. The nature of ice fog has been well defined (2). The main objection to this cold weather phenomenon is that it severely restricts visibility during abnormally difficult driving conditions. It limits commerce by closing airports and increasing automobile traffic accident rates. During ice fog conditions there are often thermal inversions which trap the fog near the ground.

Ice fog is a winter phenomenon typical of inhabited Arctic regions. It is composed of minute ice crystals that are produced when water vapor is released in ambient air that is too cold to hold it in solution. The water vapor separates into a liquid or solid phase. If cold enough, it will solidify into very small ice crystals which seem to hang in the air.

As the urban population in Alaska has increased, this fog has created serious problems for the people who attempt to live comfortably in this climate. Ice fog, capped by atmospheric thermal inversions, is known to increase the ambient levels of other pollutants such as lead compounds and toxic gases, including nitrogen and sulfur oxides, aldehydes, and halogenic acids.

Carbon monoxide is a major air pollutant in Fairbanks, but it is not directly related to ice fog. The higher levels of carbon monoxide are caused by thermal inversions that start at ground level. However, ice fog is caused by low temperatures not thermal inversions. When dense ice fog is present the thermal inversions are usually the strongest at the top of the ice fog layer. Therefore, because of the larger dilution volume under the inversion, the higher levels of carbon monoxide are not necessarily present during ice fog. Carbon monoxide is a known health hazard; the others are potential health hazards. These pollutants in the Fairbanks area have been measured by other investigators (1, 3, 4).

There are three major sources of ice fog. Ranked in decreasing order of their vision obscuring effect to the subarctic city resident, they are:

1. automobile and truck exhausts,
2. open water surfaces (such as cooling ponds), and
3. exhaust gases from heating and electrical power plants.

The relative impact of each source depends upon the individual point of view. To persons downwind from a cooling pond, the pond appears as the most important source. But, when motoring in heavy traffic, automobiles appear to be the most important source.

In the combustion sources 1 and 3 above, the water vapor is created by the oxidation of the hydrogen in the hydrocarbon fuels (gasoline, fuel oil, and coal). In the case of soft coal, much of the water vapor comes from hydrocarbon oxygenates and trapped moisture.

During winter, waste heat from power plants prevents total freeze over of the Chena River and the Fort Wainwright cooling pond. These open waters yield considerable ice fog due to high evaporation rates.

The ice fog created by home furnaces is usually injected into the atmosphere at heights of 3 to 5 meters above ground. This contribution to reduced visibility is omnipresent rather than concentrated at any one locale. Ice fog particles in the larger coal-fired power plant plumes tend to increase their overall density, thus dragging some of the toxic combustion products into the lower air layers as they settle. Because thermal inversions inhibit vertical mixing they, at times, limit some of this plume fallback into the immediate area. The extremely stable air in thermal inversions also severely limits plume dilution by dispersion. However, if the ice fog layer is deeper than the plume height, the plume is usually trapped in the ice fog thus raising the ground level sulfur dioxide concentration. To the work-a-day commuter the most significant source of ice fog is automobiles and trucks. Vehicles emit ice fog along the road network and the result is greatly reduced driver visibility. The ice fog is usually much denser at intersections. This reduced visibility in turn forces the operators to drive so slowly that increased fuel consumption and more ice fog result. The only compensating effect is that most people try to limit their driving during the extreme cold periods when ice fog is very dense.

During the four coldest months consumption of petroleum products in the Fairbanks area is about half fuel oil and half gasoline. Therefore, the ice fog contributions from heating and mobile sources are roughly equal. Together with cooling ponds they comprise most of the ground level ice fog problem.

Ice fog is not as much of a problem in the other 49 states. However, this research performed in the Fairbanks, Alaska area is applicable to any cold region where water vapor emissions are a problem.

In recognition of the severity of the ice fog problem, the U.S. Environmental Protection Agency's (EPA) Arctic Environmental Research Station (AERS) decided to apply the resources ot its Technology Research Branch. Others have done an admirable job in defining the problem (2). But what is needed now is the development and demonstration of some effective, and low-cost, control hardware.

Before discussing control methods, it is interesting to look at the relative ice fog emissions for the various fuel types. These emissions are listed in Table 1, which is extrapolated from reference (5).

Table 1. EFFECT OF FUEL ECONOMY ON AUTOMOTIVE ICE FOG (H_2O) EMISSIONS

FUEL	ASSUMED MILEAGE $\frac{km}{liter}$	$(\frac{mi}{gal})$	RESULTANT EMISSIONS $\frac{g\ H_2O}{km}$	$(\frac{oz\ H_2O}{mi})$
Diesel (Fuel Oil)	11	(26)	90	(5.1)
Gasoline (Small Vehicle)	11	(26)	79	(4.5)
Gasoline (Standard vehicle)	6.8	(16)	130	(7.4)
Propane	4.7	(11)	180	(10)

The three most common types of automotive fuels used in the Fairbanks area are propane and gasoline for the spark ignition engine and fuel oil for the diesel engine. The ice fog emission is the water vapor emission in grams per kilometer (ounces per mile). Note that the emissions are directly related to the fuel economy. In generating this table a gasoline fuel economy of 6.8 kilometers per liter (16 mi/gal) for a standard size automobile was assumed. For propane the same motive energy requirement (Joules/mi) was used resulting in 4.7 km/l (11 mi/gal). In the above case propane would emit the most ice fog, and diesel the least. Initially, the diesel powered vehicle was assumed to be 48 percent more efficient than gasoline. Recently some of the newer, smaller gasoline vehicles have mileages comparable to the larger diesels--11 km/l (26 mi/gal). In those cases, the gasoline vehicle would emit less ice fog than the diesel. This is because fuel oil contains more hydrogen per gallon than does gasoline. Fuel economy as it relates to gross vehicle weight or passenger comfort is not considered here.

In comparing emissions for any one fuel type, the water vapor (ice fog emission) is directly related to fuel economy. For example, a vehicle yielding 17 km/l (40 mi/gal) will emit only half as much ice fog as a vehicle yielding 8.5 km/l (20 mi/gal) and one-quarter as much ice fog as a vehicle attaining only 4.3 km/l (10 mi/gal).

SCOPE

There are several alternate ways to reduce automotive created ice fog without applying controls on individual vehicles. These include pooling, buses and/or electric vehicles. If any mixed application of the three above methods would result in fewer hydrocarbon powered vehicles on the streets, then the automotive-generated, on-the-road ice fog would be reduced proportionally. However, more electric power plant ice fog would result if electric powered cars come into general use. Electric automobiles use gas heaters to keep the occupants warm. Although these heaters emit ice fog the amount is much less than for gasoline powered automobiles, and it can be controlled by methods similar to those used for other automobile ice fog problems.

The AERS research effort on automotive ice fog control is the subject of this paper.

One method of controlling ice fog would be to allow the ice fog to form at a certain distance from the tail pipe, then trap it as particles. These particles could be removed with large filter assemblies such as those used in heating ventilating ducts to remove dust from air. Research has shown that in some cases electrostatic precipitation will work (2). Electrostatic precipitators are common fly ash control devices on large coal-burning power plants. The major problem with these methods would be the requirement for equipment to mix the exhaust gas with the cold air to first form the ice fog and then to capture it on filters or electrostatic precipitators. The equipment would have to be sized to handle more than 28 cubic meters per minute (1000 CFM) at a pressure drop of less than 0.25 cm (0.1 in) water column. In this case, the filtration plenum would probably be larger than the vehicle which created the exhaust. Therefore, this method would not seem to be too practical.

It is easier to limit water vapor emission than to attempt to clean up the resultant ice fog after it has formed. Two major methods of removing vapor from a gas stream are: (1) using a hydroscopic media (desiccant) such as glycol to absorb the water, or (2) cooling the gas stream below the dew point to condense out part of the water vapor. The desiccant, which may also be in a solid form (such as silica gel), is usually more effective in gas drying (dehydration) since dew points down to -40°C (-40°F) are attainable. The lower the dew point, the dryer the gas. However, the desiccant requires a contactor to absorb the water and a stripper to remove water (regenerate) before reuse. Also, exhaust contaminations such as lead compounds, soot, and mineral acids may contaminate and/or coat the desiccant.

To avoid the above complexity it was decided to concentrate on a cooler-condenser type of heat exchanger and to control the exhaust gas outlet temperature to a range of 1.7 to 4°C (35 to 40°F) to prevent freezing. Even at this high an outlet temperature, over 94 percent of the water vapor is condensed out. Figure 1 depicts three curves showing the percent of water vapor condensed for the various automotive fuel types. The calculations for these curves are in Appendix B.

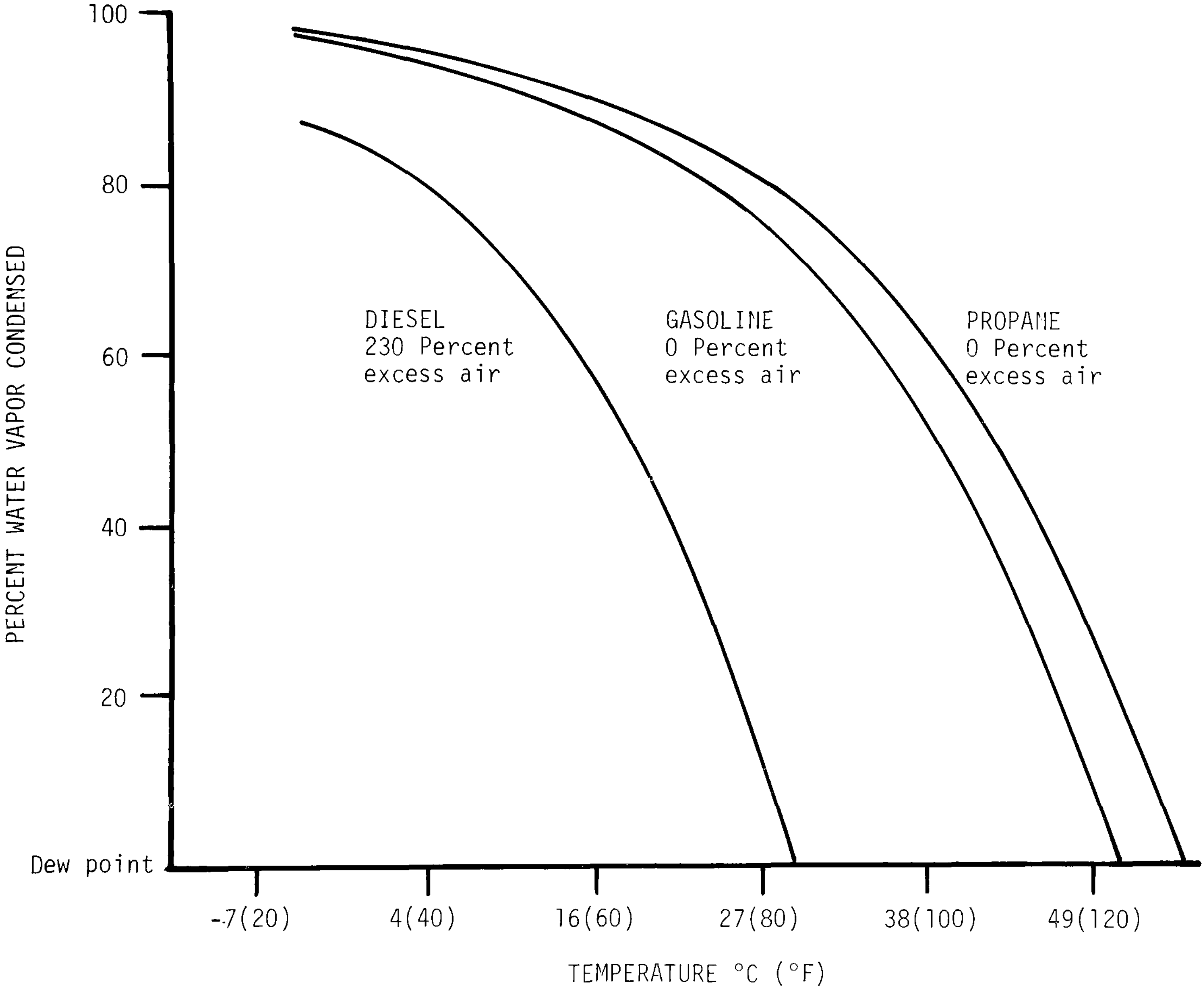


Figure 1. Condensation curves for three automotive fuels.

Because these heat exchangers cool the exhaust and condense the water vapor they are called cooler-condensers. This technique has proven successful for ice fog control from oil-fired boiler stacks (6). In the past there have been private innovators who have assembled cooler-condensers and used them successfully. The U. S. Army Cold Regions Research Engineering Laboratory is the only organization which has reported on its device (7).

Because of high local interest in the automotive ice fog problem, it was decided to support several local research contracts and to have a limited inhouse (AERS) effort during 1975. These efforts resulted in the design, construction, installation, and demonstration of nine different ice fog cooler-condensers on nine different automobiles and light duty trucks. Four were constructed by AERS and five were constructed by local engineering contractors.

SECTION 4

DESCRIPTION OF ICE FOG CONTROL DEVICE INSTALLATIONS

TEST METHODS AND INSTRUMENTS

Methods

In this research effort it was decided to build and install the cooler-condensers and evaluate them by road testing. Ice fog removal efficiency is assumed to be the same as water vapor removal efficiency which is the percent of water vapor condensed in Figure 1. Therefore, the cooler-condenser efficiency could be determined by simply measuring outlet temperature.

The road testing was performed at constant speeds of 32 and 64 km/h (20 and 40 mi/h). It was intended that the cooler-condensers be sized mainly to control ice fog in the urban-suburban areas since that is where the problem is concentrated. In these areas the speed limits are generally limited to less than 64 km/h (40 mi/h). To control ice fog emissions at higher speeds would require much larger cooler-condensers because the exhaust heat to be removed increases exponentially with vehicle speed.

Since no one drives in urban areas at a constant speed, cooler-condenser performance during a drive through town was also evaluated.

Dial Thermometers and Thermocouples

Temperature data were used to evaluate the heat exchanger (cooler-condenser) performance. Reliable temperature data were easily obtained by installing thermocouples in the exhaust gas stream, both upstream and downstream from the heat exchangers. For the higher inlet temperatures a hole was drilled at the exhanger inlet and a chromel-alumel thermocouple was inserted. To measure the low outlet temperatures an iron-constantan, thermocouple was inserted into the coalescer. Lead wires were run into the cab compartment and connected to a pair of Leeds and Northrup dial potentiometers.

To measure back pressure a hole was drilled in the exhaust pipe between the engine and the heat exchanger. A 1/8 inch N.P.S. pipe nipple was welded in with a plastic tube running into the cab to connect to a Bourdon tube pressure gauge calibrated in inches of water.

Ambient air temperatures were measured with a dial thermometer secured through an open window. The stem extended at least 15 cm (6 in.) from the body of the vehicle. In some cases the National Weather Service temperatures were used since their thermometers were within a few miles of the test road. Each interval lasted between one and two minutes to allow the changing temperatures to stabilize with respect to the thermal-kinetics of the heat exchange system.

Readings were first recorded in a log book. Later it became more feasible to use a portable tape recorder in the vehicle and transcribe the data in the office.

All measurements were in English units which in this report are in parenthesis following their equivalent metric units. Therefore, it should be realized that both 0°F and 1°F are -18°C and nominal units such as 1/2 inch EMT for example are rounded off to 1 cm EMT.

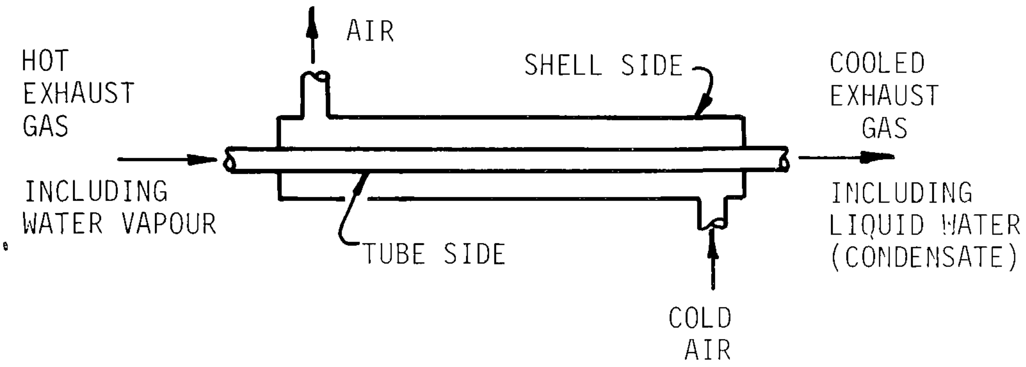
Fuel Flow Rate Meters

Fuel consumption tests were performed to determine the effect of increased back pressure upon fuel economy. These tests were conducted with the heat exchanger connected and the normal exhaust plugged. The test was then repeated with the heat exchanger plugged and normal exhaust unplugged. A constant speed of 80 km/h (50 mi/h) was maintained over a 33 km (20 mi) level section of the Richardson Highway. A stop watch was used to determine exact speed from mileage as indicated by an odometer. Exhaust back pressure was monitored and found to be constant on the level stretches.

Two fuel meters were installed in the gas line in series to check comparative precision of each. A Columbia system meter was first, followed by a Kent-Moore meter. The Columbia systems meter recorded only half the fuel actually consumed and its figures were disregarded. It was later found to have excessive internal wear.

HEAT EXHANGER DEFINITION

A heat exchanger is a device which allows heat to be passed from one fluid to another without permitting the fluids to mix. A metal wall tube usually contains one of the fluids and allows only heat to pass through. Examples of heat exchangers are automobile radiators, refrigerator condenser coils, and combustion chambers of some furnaces. A tube and shell heat exchanger can be diagramed as such:



Heat is transferred from the hot fluid, tube side in this case, through the tube wall to the shell side fluid. If there is considerable film resistance to heat transfer, the tube surface may be extended by the use of fins. The fins are on the side with the highest film resistance. If the fluids are pumped through the exchanger then it is called forced convection; if not, it is free convection. The amount of heat (BTU or Kcal) that is transferred is called the duty of the heat exchanger.

The automobile radiator is a forced convection shell-less heat exchanger in which hot antifreeze solution is pumped through the externally finned tubes. For a discussion of heat exchanger sizing techniques, see Appendix C.

Twelve of the thirteen ice fog control devices investigated here were cooler-condensers, which are heat exchangers that could cool a vehicle's exhaust enough to condense out the combustion created water vapor. This vapor, when mixed with air at sub-zero temperatures, becomes ice fog. The other control technique was an exhaust dilutor-ambient air heater. This discussion describes thirteen such devices which were installed on nine vehicles. Eight of the control devices were constructed and/or modified by the authors. Two different types were evaluated on each of the following four vehicles equipped with the following engine cubic inch displacement (CID):

(1) Metal coil flex hose cooler-condenser, first, and combination tube cooler-condenser second, on a 1968 Chevrolet Carryall (4X2) (utility vehicle), 250 CID 6 cylinder.
(2) Brazed radiator cooler-condenser first, and exhaust dilutor-ambient air heater second, on a 1971 GMC Jimmy (4X4) (utility vehicle), 250 CID 6 cylinder.

(3) Brazed radiator cooler-condenser first, and single pass stainless steel cooler-condenser second, on a 1974 Chevrolet Nova (Sedan), 250 CID 6 cylinder.
(4) Modified finned oil cooler cooler-condenser first, and a four pass stainless steel cooler-condenser second, on a 1967 Mercedes Benz (D-200), 135 CID 4 cylinder.

The other five installations were performed under contract by the following local engineering outfits. They were:

(5) Fan tube cooler-condenser on an Arctic Studies Group 1970 Volvo 144S 4 cylinder.
(6) Finned copper tubing cooler-condenser on an AE Research, Inc. 1974 Datsun, B-210, 1300 cc 4 cylinder.
(7) Liquid cooled, cooler-condenser on a H and S Research 1968 Jeep Wagoneer, 6 cylinder.
(8) Finned pipe cooler-condenser on a Scarborough & Associates 1968 Chevrolet Carryall (4X4), 307 CID V8 cylinder.
(9) Louvered shell cooler-condenser on a Simplex-Standard 1968 IHC Scout, 266 CID V8 cylinder.

All installations were designed to demonstrate the effectiveness and practicability of an ice fog removal device on an in-service automobile.

All contractors were issued guidelines by the USEPA Arctic Environmental Research Station which specified certain criteria for the scope of their project. Each contractor indicated what type of vehicle they had access to, its engine displacement, etc. In November, 1974, the contracts were awarded. By March 31, 1975, the demonstration of an ice fog removal device was required on an in-service vehicle. By May 1, 1975, a written report covering design, material cost, installation time, data obtained, and the practicality and anticipated service life of the unit was required. Photographic evidence of the system at work under sub-zero weather conditions was also required.

A short description of each device and its apparent performance follows. A more detailed description of each device is in Appendix A.

DEVICES EVALUATED BY THE ARCTIC ENVIRONMENTAL RESEARCH STATION

Metal Coil Flexhose Cooler-Condenser

The first attempts to find a solution to the Fairbanks ice fog problem were performed on a 1968 Chevrolet 1/2 ton Carryall (4X2). Its muffler was removed and 46 m (150 ft.) of flexible exhaust hose were coiled beneath the vehicle. Flexhose was chosen because it was easy to install and would not present any back pressure problems. The system proved that the exhaust temperature could be reduced to a low enough point so little or no visible ice fog was being emitted from the vehicle. Even though the hose appeared to be filled with frost there was no noticeable power reduction from back pressure. The system was only temporary and corrosion weakened the coils after two winters' use.

Combination Tube Cooler-Condenser

In the winter of 1975-76 the metal coil flexhose was removed and two conventional steel mufflers were mounted in series in the exhaust pipe. A cooler-condenser fabricated from 1 cm (1/2 in.) electrical metallic tubing was mounted between the radiator and fan. In conjunction, a section of flexhose was mounted in front of the radiator in a switch back configuration. The exhaust was directed through the mufflers, through the flexhose then into the cooler-condenser. The outlet was extended to the center of the vehicle and condensate distributed against the vehicle's drive shaft. There was no visible ice fog at any temperature at different speeds. Although this setup was over-sized intentionally, it demonstrated that all visible traces of ice fog could be eliminated. There were drawbacks such as excessive build up of condensate during long periods of idling which, if not drained out (while still in the liquid state), would freeze the entire system closed, preventing the engine from operating.

Brazed Radiator Cooler-Condenser

Late in 1974 the 1971 GMC Jimmy (4X4) was the first vehicle on which a compact cooler-condenser installation was attempted. Its first cooler-condenser was a small radiator in which solder joints were replaced with brazing alloy It was connected in lieu of the muffler It was mounted between the drive shaft and the frame. It was quite successful at idle; but exhaust temperatures above 1000°F (approx. 540°C) melted joints at the inlet header, creating excessive noise and high temperatures on the floorboards.

Free Convection-Finned Oil Cooler Cooler-Condenser

Next, early in 1975, a cooler-condenser was assembled from ten 100 cm (40 in.) lengths of 1 cm (1/2 in.) electrical metallic tubing (EMT). It lacked sufficient surface, so one half of a mobile oil cooler (Young Radiator Company MOC #6) was mounted in series. Both heat exchangers were mounted under the Jimmy between the drive shaft and frame channels; the EMT on the passenger side, and the one half MOC #6 on the driver's side. At an ambient temperature of -18°C (-1°F) this assembly would condense out 55 percent of the water vapor at idle and none at 64 km/h (40 mi/h).

Exhaust Dilutor Ambient Air Heater

Because of the relative ineffectiveness of the large surface heat exchanger mounted under the Jimmy, it was decided to try a different approach during the winter of 1975-76. This approach is based upon the principle that heating cold ambient air will increase its ability to accept water vapor without forming ice fog. The hardware involved consisted of 8 m (26 ft.) of 5 cm (2 in.) perforated spiral wound flexible metal hose connected at the tail pipe. This hose was wired to the frame channels, behind the transfer case and under the rear bumper. It functioned as a heat exchanger and exhaust gas distributor first by heating the ambient air so it could take more water vapor into solution, thereby dispersing the moist exhaust gas into the heated air. The cooler section of the metal hose also condensed some of the exhaust water vapor. With this setup the trail of visible exhaust was about half as long as without.

Because most of the exhaust was released under the floorboards, some carbon monoxide leaked through. Therefore, this system is not recommended when occupants must ride in the contaminated airspace. However, it would be quite satisfactory under the open cargo bed of a truck.

Brazed Radiator Cooler-Condenser

Another small radiator was prepared for high temperatures by brazing its seams. It was then mounted between the radiator and the grill on the 1974 Chevy Nova Sedan where the ambient air would flow over the cooling fins. In conjunction, a coalescer was fabricated out of a 61 cm (2 ft) length of 10 cm (4 in) stove pipe. An 8 cm (3 in) thick plug of furnace air filter was placed inside the pipe and the outlet exhaust directed to flow through it. Minute drops of water mist impinged on the fiberglass which caused them to coalesce and run off in a liquid stream. Condensate freezing in the radiator tubes and resultant excessive back pressure was the main problem with this application. However, at -29°C (-20°F) visible ice fog was negligible.

Single Pass Stainless Steel Cooler-Condenser

For the winter of 1975-76 a heat exchanger was built from 1 cm (1/2 in.) stainless steel tubing and mounted on the front of the Nova between the radiator and grill. Metal flexhose was used to pipe the exhaust into and out of the exchanger. A coalescer was mounted on the rear bumper and thermocouples inserted at the inlet and the outlet. Approximately 85 percent of the exhaust water vapor was condensed at temperatures below -25°C (-13°F). The addition of chains inside each tube increased internal surface area and acted as gas turbulators but did not significantly increase back pressure on the engine.

Modified Finned Oil Cooler, Cooler-Condenser

During the first winter (1974-75), a mobile oil cooler was mounted in front of the radiator on a 1967 Mercedes Benz Sedan for evaluation with a diesel engine. The exhaust was piped directly from the engine's manifold. Excessive back pressure required the removal of internal tubulators from the oil cooler. Visible ice fog diminished within a foot of the vehicle at temperatures below -18°C (0°F). One problem was the smell of exhaust fumes in the cab compartment. This could be eliminated by ensuring a leak proof mounting and extending the cooler-condenser outlet past the passenger compartment.

Four Pass Stainless Steel Cooler-Condenser

For the winter of 1975-76, a cooler-condenser with 1 cm (1/2 in.) stainless steel tubes was built and installed in place of the mobile oil cooler on the Mercedes diesel. It was designed from calculations using test data from the previous season. The problem of excessive back pressure on a diesel engine was taken into account but internal chains in the last tube pass presented too much of an obstruction for the large volume of excess air required by a diesel engine. This is not a problem for cooler-condensers used on gasoline engines. This stainless steel cooler-condenser effectively removed 70 percent of any visible ice fog. Under heavy acceleration a visible mist plume was forced out through the coalescer. However, the mist was too heavy to remain in the atmosphere and was not considered to be troublesome ice fog.

COOLER-CONDENSERS EVALUATED BY PRIVATE CONTRACTORS

Fan Tube Cooler-Condenser

Students in the University of Alaska mechanical engineering program designed and fabricated an air cooled cooler-condenser for a 1970 Volvo Sedan using 16 pieces of 1 1/2 in. X 15 in. EMT tubes enclosed in a rectangular sheet metal box (8). The exhaust gases circulated around the outside of the tubes. The device was mounted on the rear bumper. Exhaust heat was removed by cold ambient air drawn through the tubes to further reduce its relative humidity. This device prevented visible ice fog from being released into the atmosphere. Material costs were quoted at approximately $50.00 with an anticipated life of no more than three seasons.

Finned Copper Tubing Cooler-Condenser

Since the engine displacement of a 1974 Datsun Sedan was small, a manifold style cooler-condenser was fabricated from the standard copper tubing, aluminum-finned baseboard heater pipe (9). Three sections, each 91 cm (3 ft.) long, were connected in parallel and mounted under the rear of this vehicle at a slight angle to drain condensate. At idle some aerosol fog was visible. At higher speeds the greater fuel consumption increased the heat exchanger load and visible ice fog increased accordingly. To function more effectively, increased surface would be required.

Liquid Cooled Cooler-Condenser

A liquid cooled condenser rather than an air cooled type was designed by H & S Researchers. It consisted of an enclosed radiator mounted on the front of a 1968 Jeep Wagoneer (10). The coolant antifreeze solution was connected in series with the vehicle's engine cooling system. Exhaust gases were baffled back and forth through a radiator encased in a sheet metal box. The outgoing exhaust was directed at the vehicle's radiator to reevaporate mist droplets not removed in the condenser. Although visible ice fog was reduced to a minimum, this last step was undesirable since carbon monixide gas was drawn into the vehicle's cab. Safety being prerequisite, the unit was relocated to the rear of the vehicle. This made it inconvenient to use the engine's cooling system. A second radiator was incorporated using a 12 volt DC pump to circulate the antifreeze. Warm weather necessitated testing in a cold cell laboratory. No fog was visible from the mock-up.

Finned Pipe Cooler-Condenser

Scarborough & Associates built an air cooled cooler-condenser using a 1.3 m (4 ft.) section of 5 cm (2 in.) aluminuim pipe with large 15 cm X 15 cm (6 X 6 in.) fins attached to the outside (11). A spiral strip of steel inserted inside acted as a gas turbulator. It was mounted in series with the muffler under a 1968 Chevy Carryall (4X4). Effectiveness was limited because of inadequate heat transfer surface area.

Louvered Shell Cooler-condenser

Using no welding or machining of parts, a cooler-condenser was fabricated by Simplex-Standard Company using 1/2 in EMT encased in a sheet metal housing (12). This simplified method of construction demonstrated that a cooler-condenser could be built with limited tools. The unit was designed so that header covers could be removed for inspection and cleaning. It was mounted on the front bumper of a 1968 IHC Scout and fed by a section of flex exhaust hose extending from the tail pipe. Air flowing across the tubes could be externally controlled by a cable attached to adjustable louvers in front of the tubes. The unit reduced a substantial amount of visible water vapor at idle and was quite effective at higher speed.

SECTION 5

DEVICE PERFORMANCE AND COMPARISON

When considering the cooler-condenser type of ice fog control device the only performance criterion is the fraction of exhaust gas water vapor condensed (removed). The percent of the water vapor condensed is directly related to the outlet temperature (assuming constant pressure) as shown in Figures 1 and 2. The calculations are in Appendix B. Therefore, the cooler-condenser (heat exchanger) performance can be measured in terms of the one most important parameter, outlet temperature. Efforts were directed toward reducing the temperature enough to condense out 95 percent of the water vapor. From Figure 1 that temperature must be 4°C (40°F) or less for gasoline. Because of excessive ice formation it is not practical to condense more than 85 percent of the moisture in diesel engine exhaust.

FIRST WINTER RESULTS

The results from the first winter's (1974-75) installations will be discussed first. The coiled flexhose under the 1968 Chevy Carryall (4x2) will not be compared with the other cooler-condensers because it was too large to be considered practical and it clogged easily with frost.

The comparative performances of the remaining eight devices are tabulated in Table 2. Listed with each cooler-condenser is the vehicle on which it was installed.

In all the cooler-condensers except the one on the Jeep the exhaust heat was transferred directly to ambient air Free air convection means that the cooler-condenser was mounted parallel to the air stream or such that one tube shaded the rest from the air movement. Forced convection means that air was forced across the tubes either by vehicle movement or a fan. The forced convection condensers had to be mounted in front of the vehicle of have a fan blowing air across the tubes.

The 1968 Chevy Carryall (4x4) belonged to a contractor and had only one finned tube hanging below the vehicle, so it was called a forced convection cooler-condenser

The percent water condensed (Columns 1 and 2) was obtained by measuring the outlet temperature then reading the percent condensed off of Figure 1. Sixty-four km h (40 mi h) was chosen as the upper reasonable speed limit for sizing the cooler-condenser duty since urban traffic seldom exceeds that speed during ice fog conditions.

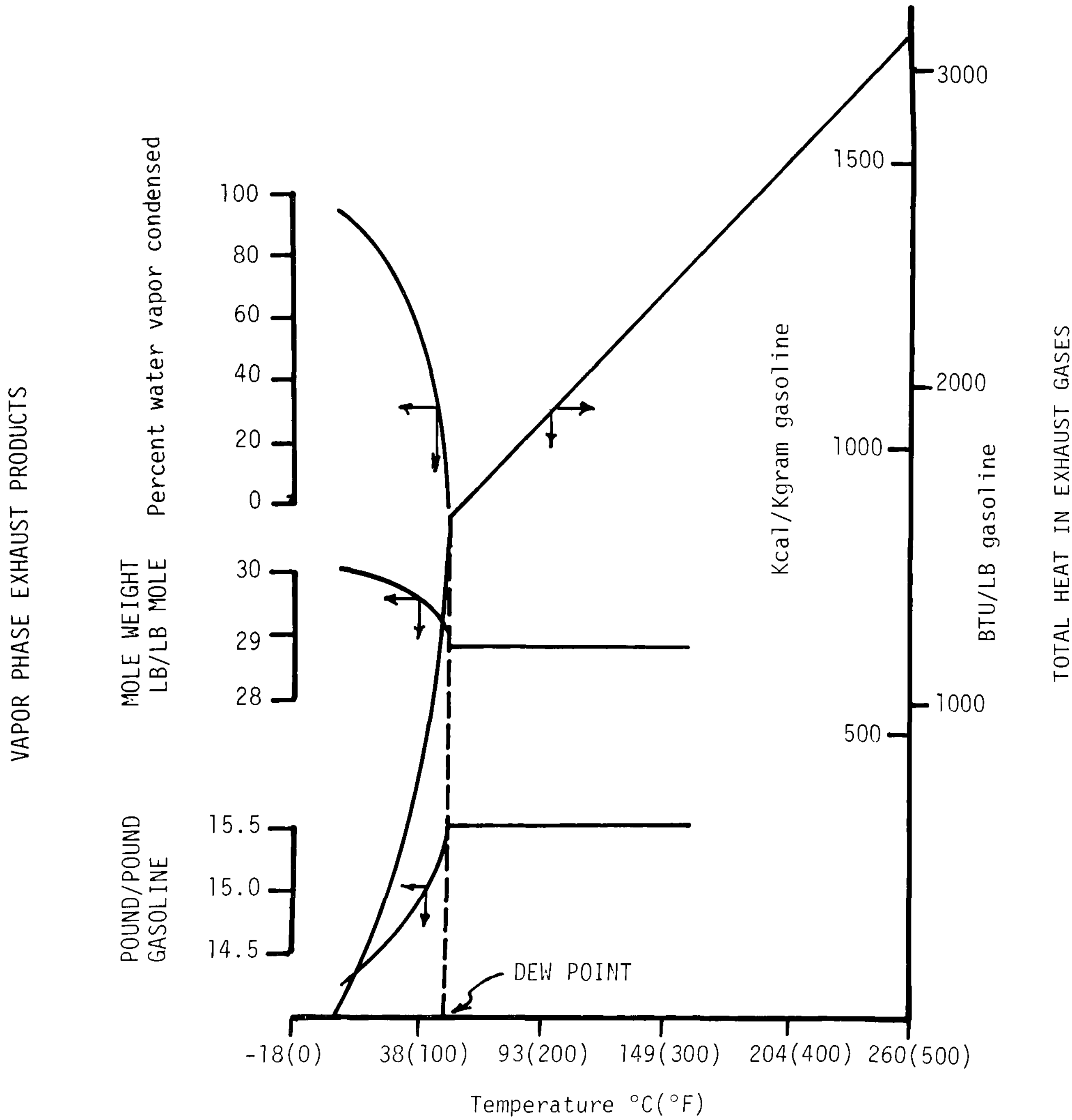


Figure 2. Exhaust gas heat content, Kcal/Kgram of gasoline (C_7H_{13}) consumed. Exhaust gas vapor mole weight and mass per mass gasoline. Percent water vapor condensed vs. temperature.

TABLE 2. FIRST WINTER PERFORMANCE OF PROTOTYPE AUTOMOTIVE ICE FOG CONTROL DEVICES

Type of Cooler-Condenser and (Vehicle)	Percent exhaust H_2O Vapor Condensed and Ambient Temperature		Heat Transfer Surface-Hot-Gas Side	Heat Transfer Coefficient kcal/m^2-h-°C (BTU/h-ft^2-°F)	Required Heat Transfer Surface at -29°C (-20°F)	Ratio of Finned to Tube Area	Exhaust System Back Pressure at 64 km/h (40 mi/h)
	at Idle	at 64 km/h (40 mi/h)					
Finned Tube, free convection (1971 GMC Jimmy 4x4)	55% -18°C (- 1°F)	0% -18°C (- 1°F)	1.1 m^2 (11.8 ft^2)	43 (8.9)	2.1 m^2 (23 ft^2)	5:1	56 cm of water
Brazed Radiator forced convection (1967 Chevy Nova Sedan)	95% -19°C (- 3°F)	83% -18°C (- 1°F)	0.72 m^2 (7.8 ft^2)	100 (21)	0.78 m^2 (8.4 ft^2)	5:1	106 cm of water
Modified finned oil cooler, forced convection (1967 Mercedes Benz D200)	25% -24°C (-12°F)	0% -24°C (-12°F)	0.21 m^2 (2.3 ft^2)	127 (26)	0.50 m^2 (5.4 ft^2)	8:1	105 cm of water
Fan Tube, forced convection (1970 Volvo Sedan 144S)	98% -18°C (- 1°F)	No Data	1.0 m^2 (11 ft^2)	No Road Test Data		1:1	Nil
Finned Copper Tubing, mixed convection (1974 Datsun B-210 Sedan)	97% -17°C (1°F)	0% -14°C (6°F)	0.33 m^2 (3.6 ft^2)	54 (11)	0.79 m^2 (8.6 ft^2)	13:1	Nil
Louvered Shell, forced Convection (1969 IH Scout 4x4)	90% -17°C (0°F)	88% -17°C (0°F)	0.93 m^2 (10 ft^2)	73 (15)	0.92 m^2 (9.9 ft^2)	1:1	35 psig (see text)
Finned Pipe, forced convection (1968 Chevy Carryall 4x4)	47% -20°C (- 5°F)	0% -20°C (- 5°F)	0.25 m^2) (2.7 ft^2)	42 (8.6)	0.48 m^2 (5.2 ft^2)	15:1	Nil
Liquid Cooled, forced convection (1968 Jeep Wagoneer)	47% -12°C (10°F)	No Data	1.75 m^2 (18.9 ft^2)	Depends on Antifreeze temperature		5:1	Nil

Mathematics detailing the method for calculating the overall heat transfer coefficient (Column 4) are presented in Appendix C. This coefficient is the transferred heat rate (duty) from Figure 2 divided by the temperature difference (log mean) and the surface area (inside hot gas side, Column 3). The coefficient was calculated from the ambient temperatures listed in Columns 1 and 2 and the inlet and outlet temperatures. The surface required to condense out 95 percent of the water vapor at -29°C (-20°F) (Column 5) ambient was then calculated using the above heat transfer coefficient (Column 4). All calculations were based upon the inside surface area since data from the Engineering Handbook (13) show the inside (exhaust gas) film as having the most resistance to heat transfer. This is true only when the outside air velocity across the outside tub surface is high enough to significantly reduce the outside film thickness. For a definition of film resistance, see Appendix C.

The ratio of the extended surface to the internal surface is listed in Column 6. The condensers that were modified radiators had ratios of 5:1 or greater. This is because they were designed for liquid in the tubes and air outside. The resistance of the liquid film to heat transfer is small when compared to gas film resistance. Therefore, to compensate, the gas film surface area is made much larger.

The only cooler-condenser that did not use ambient air to cool the exhaust was on the 1968 Jeep. This condenser was a modified automobile radiator in which cold antifreeze from the Jeep's normal radiator was used as the cooling medium. Exhaust gas was baffled through the finned area of the radiator. Because of increased duty this system would be at a disadvantage when the normal radiator isn't able to cool the antifreeze to below 5°C (40°F). Under that condition 95 percent of the exhaust water vapor could not be condensed out (Figure 1).

To find out whether or not this is a significant hindrance, a thermocouple was wired to the radiator return antifreeze hose on a 1968 Chevrolet Carryall with a 250 CID engine. Insulation was placed over the probe so that its temperature would be close to that of the antifreeze. At road speeds from idle to 64 km/h (40 mi/h) the antifreeze temperature varied from 5 to 8°C (9 to 14°F) above ambient at ambient temperatures less than -18°C (0°F). But when the vehicle was under heavy load, such as accelerating up a 10 percent grade, the return antifreeze temperature rose to 30°C (54°F) above ambient. For exhaust condensation it is only necessary to consider normal road load since ice fog is not as much of a problem in hilly terrain. Therefore, at normal road load, the antifreeze temperature is low enough for efficient condensation when the ambient temperature is below -18°C (0°F). Automobile created ice fog is generally not a problem at temperatures above -18°C (0°F).

A lower tendency for ice accumulation is a major advantage of passing exhaust gas through the shell side of a cooler-condenser. The flow area on the shell side is much larger than that through the tubes and will accomodate much more ice before blocking. Exhaust gases on the 1970 Volvo flowed through the shell side of its cooler-condenser.

A first year quantitative performance comparison among the various cooler-condensers is difficult because of lack of consistent test procedures, different ambient conditions, and lack of accurate temperature data. However, some statements can be made. All the cooler-condensers were successful in removing some of the water vapor, particularly the cooler-condenser on the Nova, Datsun, and Scout, which condensed over 90 percent of the water vapor at engine idle. Only the condensers on the Scout and Nova were effective at 64 km/h (40 mi/h). Direct cooler-condenser comparisons are made difficult because the exhaust gas into the Nova's cooler-condenser was above 260°C (550°F) while the temperature into the Scout's was less than 93°C (200°F). The Nova's exhaust was tapped before the muffler; the Scout's after. The short connecting hose from the exhaust system to the cooler-condenser on the Nova did not perform much cooling because of its small surface; compared to that of a muffler. The 0.2 to 0.4 square meters (2 to 4 square ft) muffler surface on the Scout removed considerable superheat from the exhaust.

For the heat transfer calculations, the estimated exhaust temperatures into the condenser at 64 km/h (40 mi/h) were 150°C (300°F) for the Scout and 430°C (800°F) for the Nova. Because of low inlet temperature, the cooler-condenser on the Scout was the only one adequately sized to do the job at 64 km/h (40 mi/h) at ambients of -29°C (-20°F) or less.

The effect of the type of coolant circulation on the overall heat transfer coefficient is shown in Column 4. All cooler-condensers with forced convection had heat transfer coefficients of 73 kcal/h-m^2-°C or more. The 1968 Chevy Carryall 4X4 transfer coefficient was evaluated at idle-free convection. For the other cooler-condensers with free convection, the transfer coefficients were 54 kcal/h-m^2-°C or less. The cooler-condenser on the Datsun is a mixture of forced and free convection because one tube partially shelters the rest from the wind flow.

From the above discussion, it is obvious that to be effective, the cooler-condensers must be exposed to the wind or have some other means, such as a fan, to create air turbulence. For example, without forced convection, the surface requirement on the Jimmy cooler-condenser is 2.1 square meters (23 square ft).

The superiority of the Nova cooler-condenser is further exemplified by Figure 3 which is a plot of cooler-condenser outlet temperatures vs. vehicle speed. This figure also shows that the outlet temperature may vary by as much as 19°C (35°F) at any one speed. That is because of varying cooler-condenser duty caused by slight road slope and/or imperceptible acceleration-deceleration.

The effect of engine load is further demonstrated in Figure 4 which shows that for a fixed vehicle speed, higher engine rpm places more duty on the cooler-condenser resulting in a higher outlet temperature. For a given load and vehicle speed, an engine running at a higher rpm (lower gear) will put out more waste heat because it has to overcome increased internal friction.

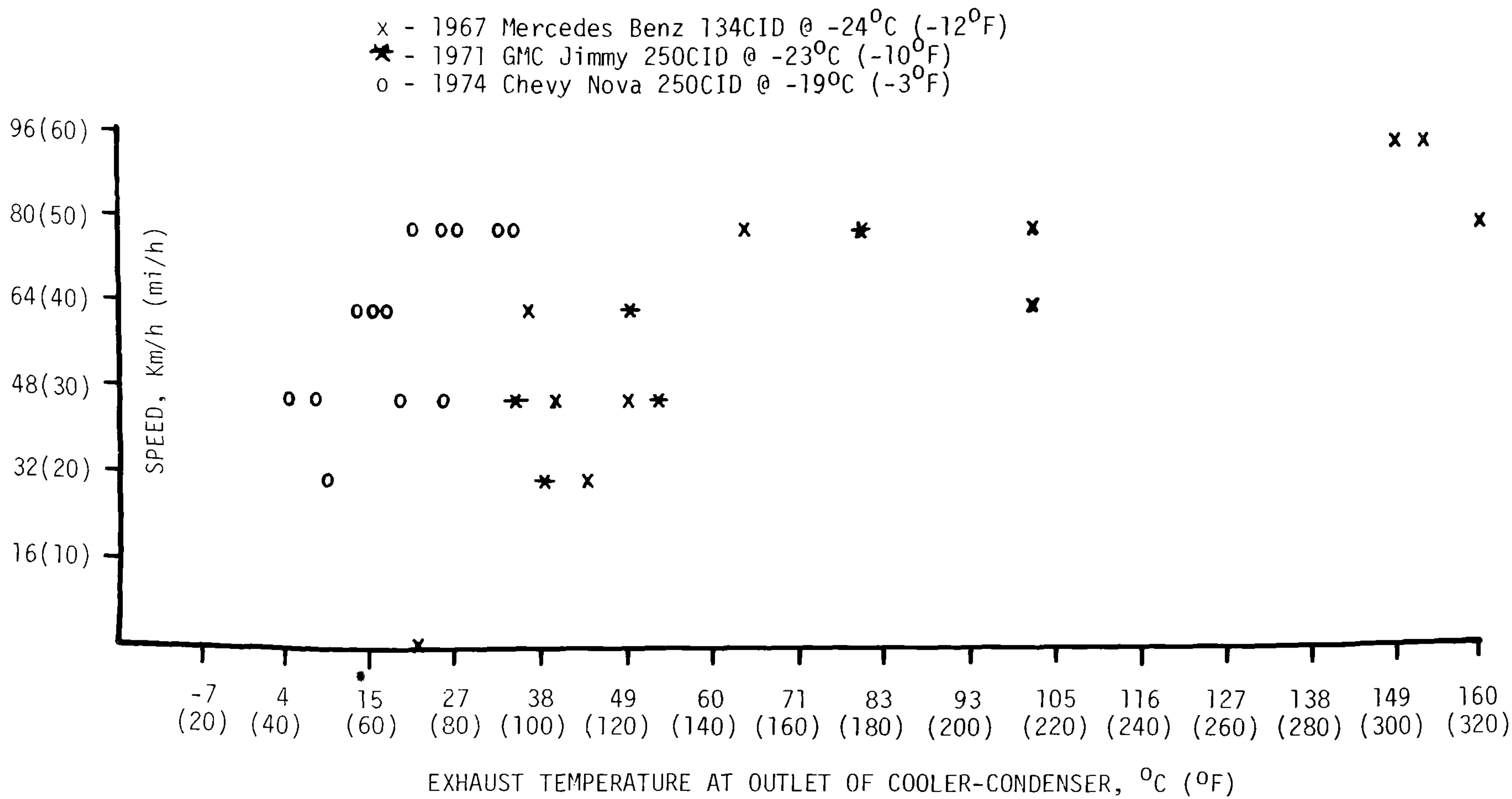


Figure 3. First winter cooler-condenser exhaust temperature ranges.

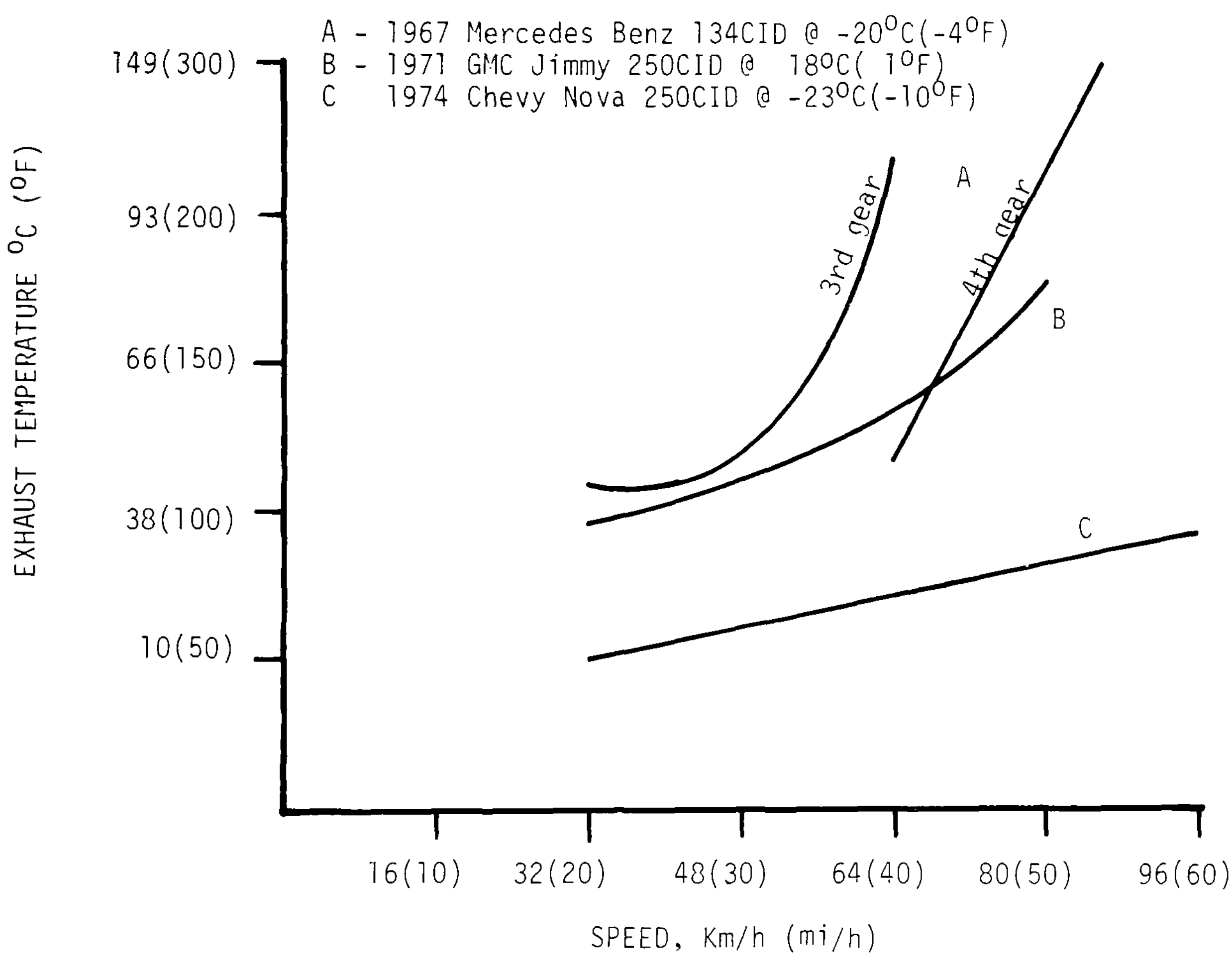


Figure 4. First winter cooler-condenser exhaust temperature

Measuring the outlet temperature and calculating the percent water vapor condensed does not always show the whole picture because it was noticed that some cooler-condensers had exhaust that was less visible than others even though both had condensed over 90 percent of the water vapor. This difference is thought to be caused by relative humidity of the ambient air. If the air is saturated (relative humidity = 100%), then any water vapor emission, no matter how small, will at low temperatures appear as ice fog. The relative humidity appears to be less during late winter then early winter. Therefore, ice fog control generally appears more effective in Februrary then in December, even though the same percent water vapor is condensed in both cases.

Mist Coalescers

Cooling the exhaust gas to well below the dew point does not ensure that all the condensed water will collect as one easily dischargeable aliquot. Some of the water vapor condenses into discrete minute droplets (mist) which when emitted appear as a dense ice fog. This problem was first noticed with the Nova and Jimmy cooler-condensers. A first attempt in removal was to direct the outlet stream against a large cold metal surface hoping the droplets would freeze to it. This was done by directing the Jimmy's cooled exhaust against a large metal skid plate. Success was limited, probably because the exhaust was not cold enough to begin with.

Next, a coalescer (Figure 5), was fabricated for the Nova cooler-condenser. This coalescer was a 61 cm (24 in.) length of 10 cm (4 in.) diameter stove pipe. A 8 cm (3 in.) thick expanded fiberglass air filter plug was placed near one end. The condenser exhaust was injected by a 4.4 cm (1-3/4 in.) hose into the other end of the stove pipe and directed toward the plug.

The injected exhaust plus some inspired ambient air passed through the coalescing fiberglass plug which captured (by impingement) the minute droplets causing them to grow until they were heavy enough to drip out of the coalescer. If for some reason the fiberglass plug should freeze solid, then the exhaust would simple exit the stove pipe at the inlet end since the annular space between the inlet hose and the stove pipe was open. The coalescer performed well; there was no longer any readily visible persistent mist during vehicle operation.

The investigators with the Volvo eliminated the mist problem by reevaporating it. They mixed the exhaust with ambient air being drawn through the tubes. On the Jeep, the mist laden exhaust was directed at the vehicle's radiator where it was reevaporated and mixed with the warm dry air. One major problem with this is the possibility of carbon monixide poisoning of occupants in the vehicles, especially if the fire wall between the engine and passenger compartments is not airtight.

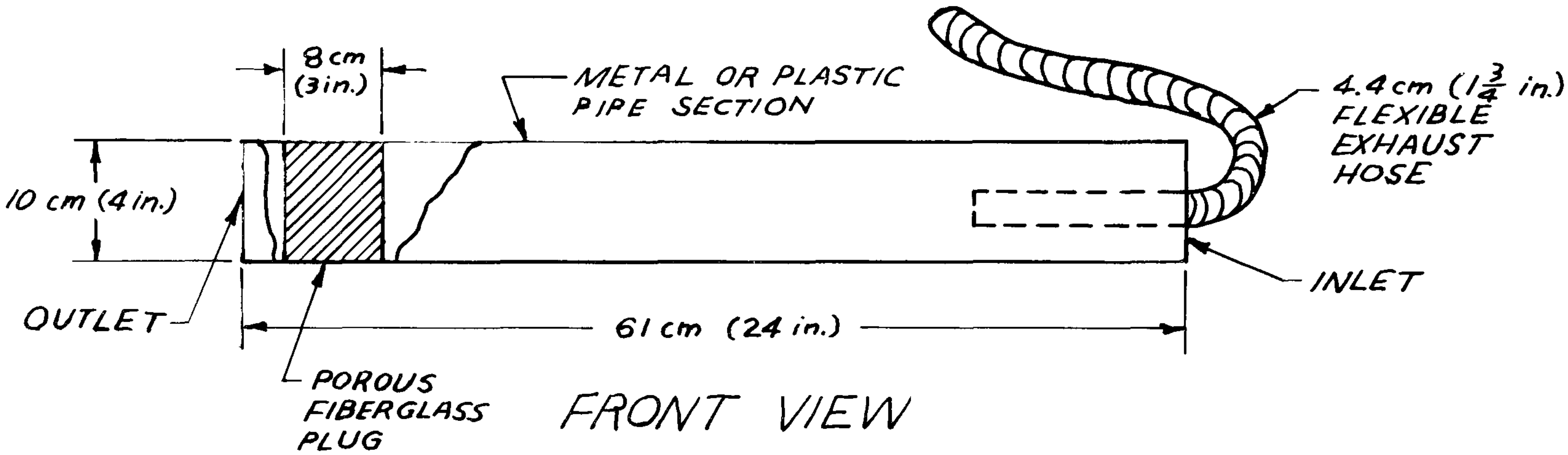


Figure 5. First mist coalescer.

In summary, some method of removing the mist that escapes the cooler-condenser must be devised. Reevaporation or coalescence will work, but re-evaporation will only add to the atmospheric water vapor, part of which will result as ice fog at some distance from the vehicle. Therefore, coalescence is the best method.

The coalescer mounting direction is critical. Mounted paralled to vehicle movement, it tends to freeze shut; mounted perpendicular there have been no problems. It is also thought that 8 or 10 cm (3 or 4 in.) plastic pipe makes a better coalescer shell since it accumulates less ice than does metal. During the second winter, coalescer lengths of 35 cm (14 in.) were found to be adequate.

SECOND WINTER RESULTS

For the winter of 1975-76, the AERS extrapolated the results from the first winter (Table 2 column 5) to size and fabricate three cooler-condensers. Later it was necessary to add more surface because the extrapolated values from the first winter proved inadequate. Also, one exhaust dilutor-heat exhanger was attached to the 1971 Jimmy.

The second winter results were analyzed in more detail because by then the field evaluation techniques had become more standardized.

Constant Speed Performance

All the constant load cooler-condenser output temperatures were measured by driving at a constant speed on the near level section of the Chena Pump Station Road, a few kilometers south of the AERS. Temperature measurements were taken going out as well as coming back. The constant speed results were usually within a few °C of each other.

After changing speeds, it would take several minutes for the cooler-condenser outlet temperatures to reach equilibrium. In most cases after the rate of temperature change dropped to less than 1°C per minute, the temperatures were recorded. In later tests it was discovered that the temperatures would slowly continue to drop by as much as 7°C after they had been recorded. For example, with the gasoline engine this error would cause the actual percent condensed to be 6 percent low for a hastily read outlet temperature of 21°C (69°F). At low temperatures the temperature error has little effect on performance (percent condensed) because the condensation curves flatten out when more than 85 percent of the water vapor is condensed.

In actual vehicle use there is no such thing as constant speed level road driving. The engine load seldom remains constant long enough for the cooler-condenser to attain equilibrium. Therefore, it is considered better to be on the conservative side when describing cooler-condenser performance. To verify actual situation performance a drive-though-town test was made.

Combination Coil/Tube-Cooler-Condenser--

The cooler-condenser on the 1968 Carryall consisted of three separate heat exchangers: two mufflers in series, then a looped length of spiral wound flexhose, followed by an EMT cooler-condenser then more flexhose. The system was evaluated with and without the two lengths of spiral wound flexhose. Each respective total heat transfer surface was 1.9 and 1.1 square meters (21 & 12 square feet).

Performance of the two systems is shown in Figure 6. At temperatures below -37°C (-35°F) and speeds below 20 km/h (12 mi/h), both systems condensed out over 90 percent of the water vapor. At -21°C (-6°F) the smaller system was not capable of condensing out any water vapor. It had inadequate surface. The total system of 1.9 square meters (21 square feet) was more than adequate since it condensed over 80 percent of the water vapor at 64 km/h (40 mi/h) at an ambient of -11°C (+12°F). Its condensation performance at lower temperatures was slightly better but never exceeded 95 percent; probably because internal ice films built up and decreased the heat transfer efficiency.

In sizing a cooler-condenser for this vehicle it appears that 1.1 square meters is not enough surface. But, 1.9 square meters will provide for greater than 80 percent condensation at speeds up to 64 km/h (40 mi/h) and ambients temperatures -11°C (+12°F) or less. The muffler, but not the connecting tailpipe surface, is included in the 1.9 square meters.

When making extrapolations to other vehicles it must be recognized that a cooler-condenser's surface is directly related to its duty. For example, a heavier vehicle with a larger engine would require a proportionally larger surface.

Single Pass Stainless Steel Cooler-Condenser--

The second winter cooler-condenser performance on the 1974 Chevy Nova is shown in Figure 7. Curves 1 through 5 were results with the front outlet coalescer mounted below the bumper on the driver's side. The total surface area including the inlet flexhose amounted to 1.3 square meters (14 square feet). It was capable of about 60 percent condensation at 64 km/h (40 mi/h) for temperatures below -19°C (-13°F). The poorest high speed performance, steepest curve, is at the lowest temperature, -42°C (-43°F). This was probably due to ice film formation. The ice film creates more resistance to the flow of heat through the tube walls. For more discussion on icing see the section on back pressures. Also note at this ambient that the idle performance exceeded 90 percent condensed. This value may be high because the ice formed at the higher speed is probably melting and increasing efficiency at idle.

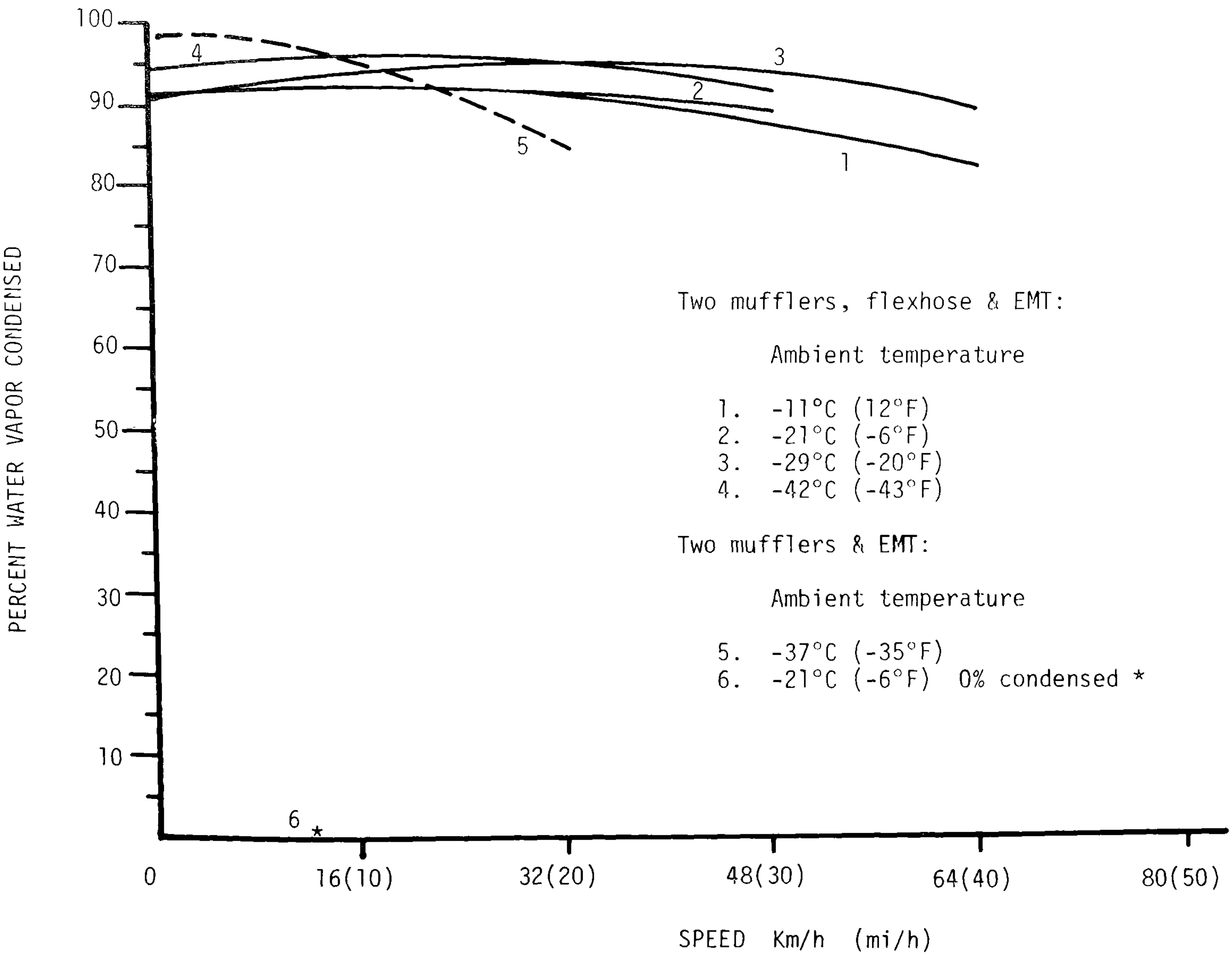


Figure 6. 1968 Chevy Carryall (4 x 2) second winter cooler-condenser performance.

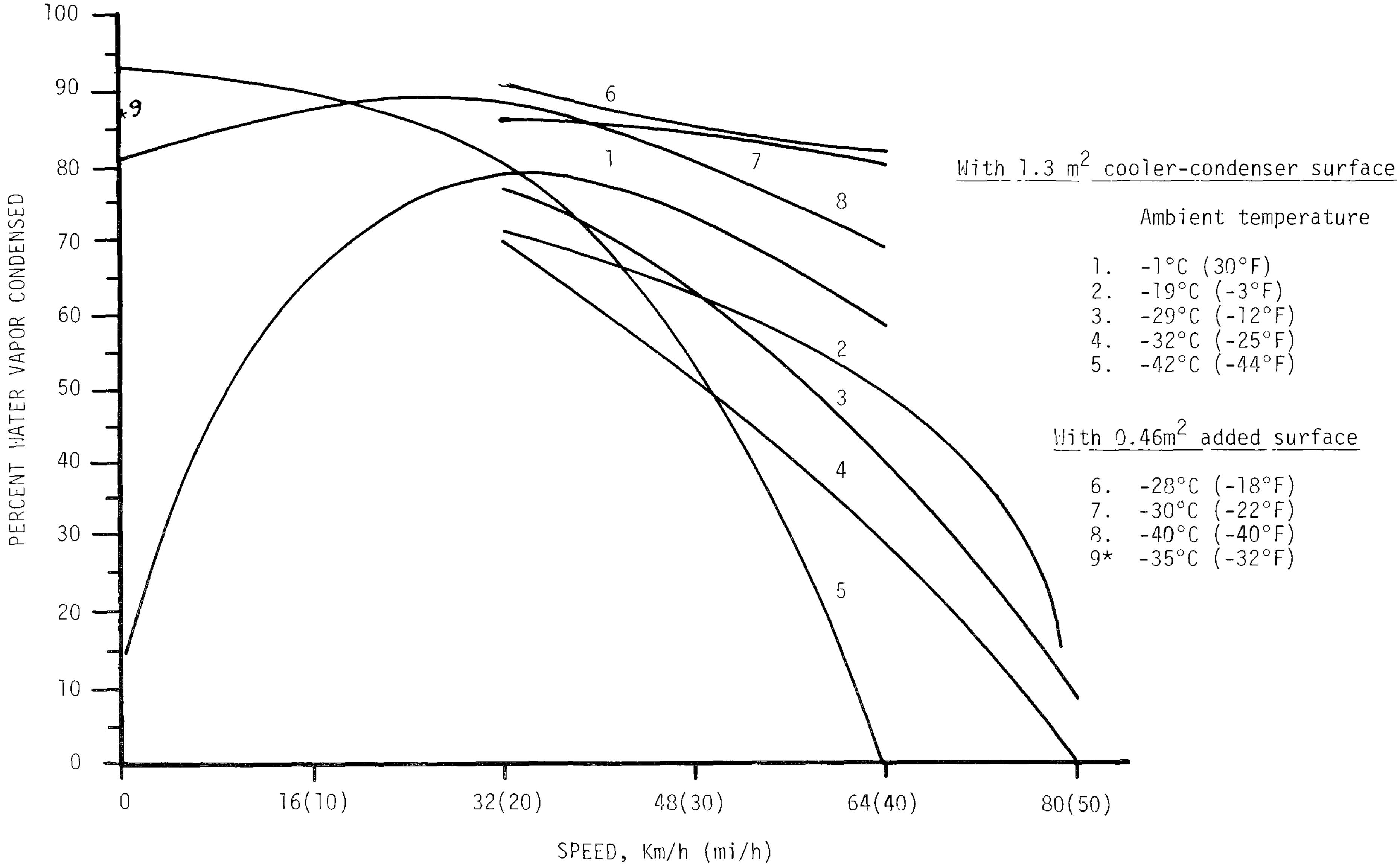


Figure 7. 1974 Chevy Nova second winter cooler-condenser performance.

When temperatures are near freezing (-1°C, 30°F) the idle performance is poor because the temperature difference across the cooler-condenser is low. But at speeds greater than 35 km/h (22 mi/h) the performance exceeds that for all the lower temperatures, probably because there is no ice film to restrict the flow of heat at the -1°C (30°F) ambient temperature.

Adding enough spiral wound flexhose to extend the outlet to the rear of the vehicle increased the cooler-condenser surface to 1.8 square meters (19 square feet). This increased surface resulted in 80 percent or more condensation at speeds less than 64 km/h (40 mi/h) for ambients -30°C (-22°F) and warmer. At -40°C (-40°F), 68 percent was condensed at 64 km/h (40 mi/h).

Because it would not hold water, the spiral wound flexhose never filled with frozen condensate.

Four Pass Stainless Steel Cooler-Condenser--

The performance of the cooler-condenser on the 1967 Mercedes Benz diesel is shown in Figure 8. Curves 1 through 6 were results when a 3.8 cm (1.5 in.) diameter spiral wound metal flexhose was used to connect the cooler-condenser outlet to the muffler inlet. Curve 7 illustrates data when the connection was made with a plastic suction hose. The spiral wound flexhose added 0.14 square meters (1.5 square feet) of heat transfer surface and, since it leaked, reduced the condensate freezing problems in the muffler and downstream. The cooler-condenser performed well only at idle. Its lower performance under load at the lower temperatures probably indicates ice formation on the inside tube walls.

The diesel engine is the only automotive type engine that operates with excess air. Its exhaust carries a larger fraction of noncondensables than a gasoline engine exhaust. The diesels exhaust must therefore be cooled to a lower temperature to achieve the same percentage condensed as a gasoline engine's exhaust. In other words, for the same percentage condensed, the diesel exhaust condensate is nearer freezing; see Figure 1.

Exhaust Dilutor-Ambient Air Heater--

The eight meters (26 ft) of perforated spiral wound flexhose connected to the tailpipe and mounted under the 1971 Jimmy functioned as an exhaust dilutor-ambient air heater. It heated ambient air so exhaust water vapor would be accepted without forming ice fog. There is also some associated exhaust condensation, but that is not the device's principal function. The exhaust is being dispersed along the hose length and diluted as its temperature drops. There is much more visible ice fog reduction than a condensation curve would show. The device performance is best displayed by the with and without photos, Figures 9 and 10. Both photos were taken within five minutes of each other at an ambient of -29°C (-20°F). In both cases the vehicle was idling about 1100 rpm. In figure 9 the dilutor-ambient air heater hose can be seen slipped over the tailpipe below the left tail light. Without the device the person standing at the left of the vehicle is almost completely obscured. The visual effect during driving is similar, but not as spectacular.

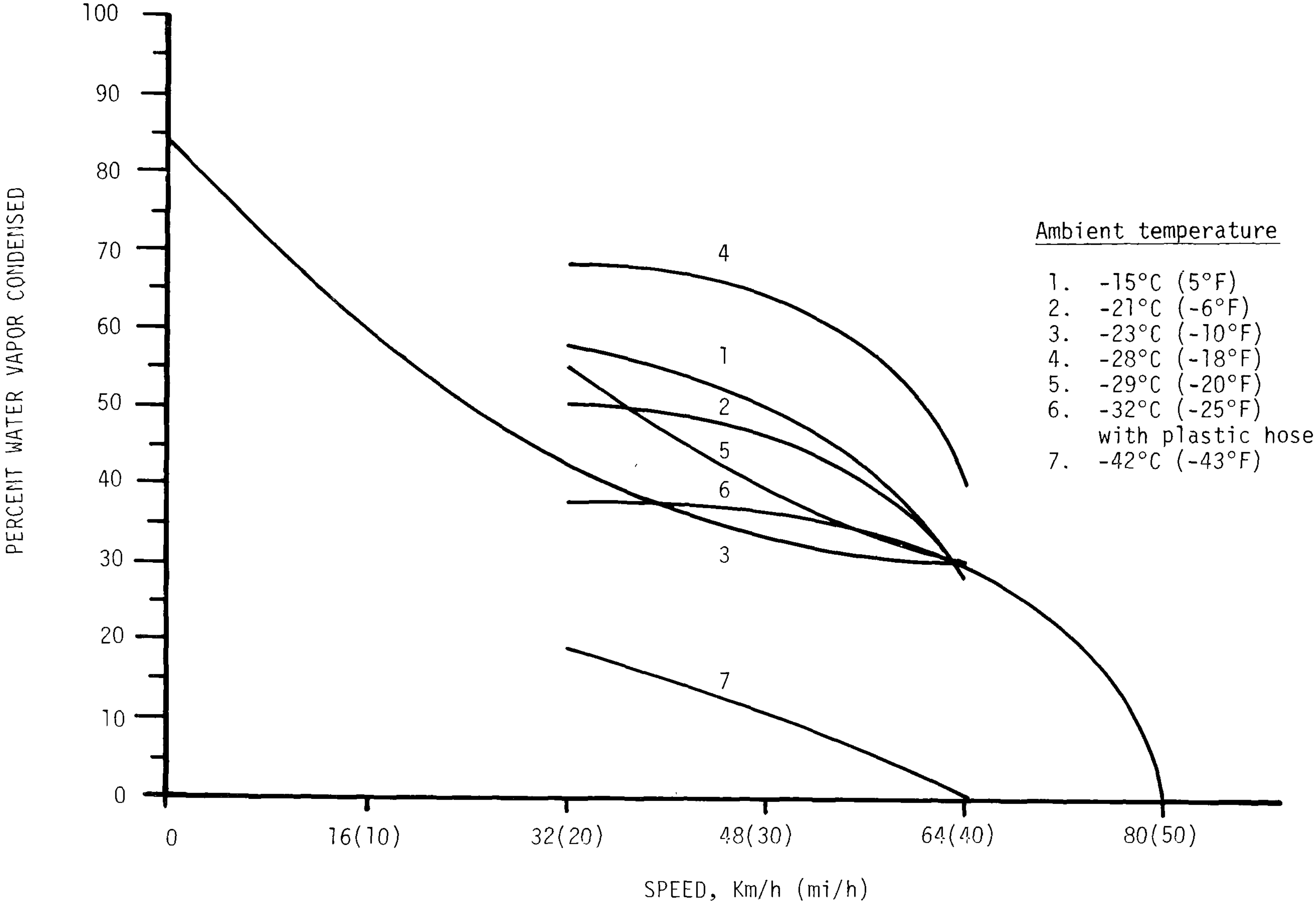


Figure 8. 1967 Mercedes Benz diesel second winter cooler-condenser performance.

Figure 9. 1971 GMC Jimmy with dilutor - ambient air heater

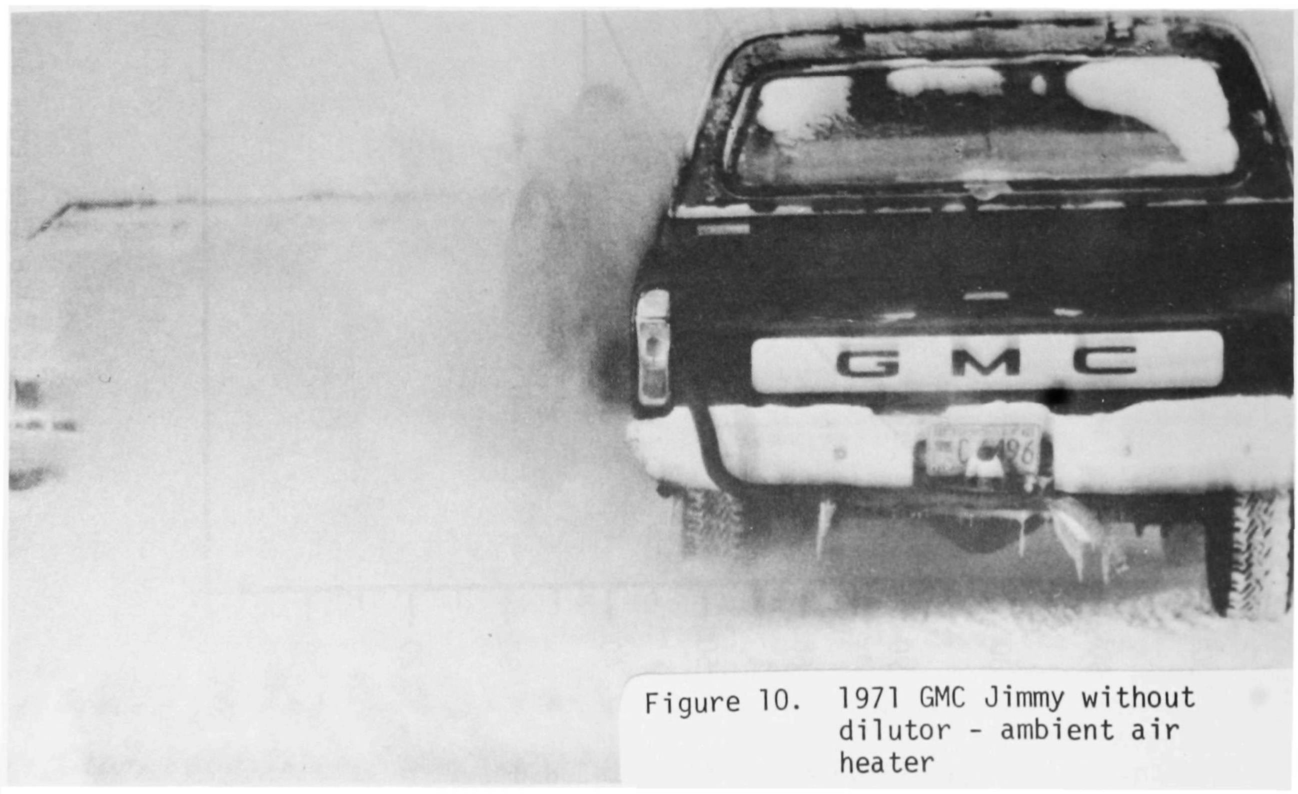


Figure 10. 1971 GMC Jimmy without dilutor - ambient air heater

Cooler-Condenser Sizing--

When comparing sizing criteria for the exhaust gas to air forced convection cooler-condenser, two different estimates were obtained. One was extrapolated from the first winter's results. The other from the Engineering Manual (13). The first winter's results indicated that about 0.93 square meters (10 square feet) of transfer surface was needed to condense out 95 percent of the water vapor at 64 km/h (40 mi/h) and -29°C (-20°F) ambient. On the other hand, calculations based upon data in the Engineering Manual yield about 1.9 square meters (21 square feet) as the required surface (Appendix C). In field tests about 1.9 square meters (21 square feet) of transfer surface was found to be required. The probable reason for this discrepancy is that the first winter's data were obtained at warmer temperatures when internal surface icing would not have reduced the overall heat transfer coefficient.

Diesels have a high percentage of non-condensables (non water) in their exhaust and they usually emit comparatively less ice fog than gasoline or propane engines (Table 1). Because of the condensate freezing problem and low water vapor exhaust concentration, it is expected that the air against exhaust gas cooler-condensers will require more surface on diesel powered vehicles. This fact is evident from the tabulated coefficients in Table 3, which is a comparison of the overall heat transfer coefficients for the second winter's cooler-condensers. Because the fuel economies were assumed, the heat transfer coefficients have an accuracy of only one significant figure.

Drive Through Town

To demonstrate the actual performance of the ice fog cooler-condensers a special test was performed by driving the 1974 Chevy Nova through town. Data were gathered under normal city driving conditions. Temperatures, speed, location and time were recorded every 15 seconds. Odometer readings were taken at different intervals so that an average town speed could be calculated.

Speed varied between 0 and 64 km/h (40 mi/h) with an average intown speed of 13 km/h (22 mi/h). Figure 11 is a graph showing percent water vapor condensed and vehicle speed vs. time. Outlet temperatures ranged between 9°C (48°F) and 36°C (97°F) with an overall average of 21°C (60°F). Ambient air temperature was 16°C (4°F) Amounts of water vapor condensed ranged from a low of 57 percent under heavy acceleration to a high of 92 percent at idle. The overall amount condensed for the total run was 81 percent. The day was clear with open roads and occasional ice patches. Traffic was moderate to heavy at the busier intersections.

Back Pressure and Tube Icing

Another important consideration is the added back pressure on the engine caused by the cooler-condensers. The Jimmy (first winter), Nova and Mercedes had their exhaust manifolds connected directly to a cooler-condenser in place

TABLE 3. SECOND WINTER COMPARISON OF OVERALL HEAT TRANSFER COEFFICIENTS AT 64 km/h (40 mi/h)

TYPE OF COOLER-CONDENSER Vehicle and Exchanger Surface m^2 (ft^2)		TEMPERATURES AMBIENT AIR °C (°F)	COOLER-CONDENSERS OUT °C (°F)	COOLER-CONDENSERS IN °C (°F)	ASSUMED FUEL ECONOMY km/l (mi/gal)	EXHAUST HEAT REMOVED kcal/h (BTU/h) in 1000's	OVERALL COEFFICIENT $U=Q/A\Delta T$ $\frac{kcal}{h\text{-}m^2\text{-}°C}$ $\left(\frac{BTU}{h\text{-}ft^2\text{-}°F}\right)$
Gasoline Engines							
Combination Coil/tube on 1968 Chevy Carryall							
0.79	(8.5)	-9(15)	16(60)	74(165)	7.7(18)	5.2(21)	90(18)
0.79	(8.5)	-29(-20)	12(53)	171(340)	7.7(18)	8.4(33)	120(25)
0.79	(8.5)	-41(-43)	7(45)	107(225)	7.7(18)	7.0(28)	120(24)
Single Pass Stainless Steel on 1974 Nova							
0.97	(10.4)	-19(-3)	34(94)	238(460)	8.5(20)	7.1(29)	60(12)
1.42	(15.3)	-29(-20)	27(80)	260(500)	8.5(20)	8.2(32)	40(8)
0.97	(10.4)	-40(-40)	51(123)	341(645)	8.5(20)	6.5(26)	30(7)
1.42	(15.3)	-40(-40)	32(90)	299(570)	8.5(20)	8.7(35)	30(7)
Diesel Engine							
Four Pass Stainless Steel on 1967 Mercedes Benz							
2.28	(24.5)	-20(-5)	24(75)	249(480)	13(30)	4.1(16)	15(3)
2.28	(24.5)	-31(-25)	22(71)	282(540)	13(30)	5.0(20)	15(3)
1.91	(20.5)	-41(-42)	27(88)	271(520)	13(30)	3.4(14)	10(2)

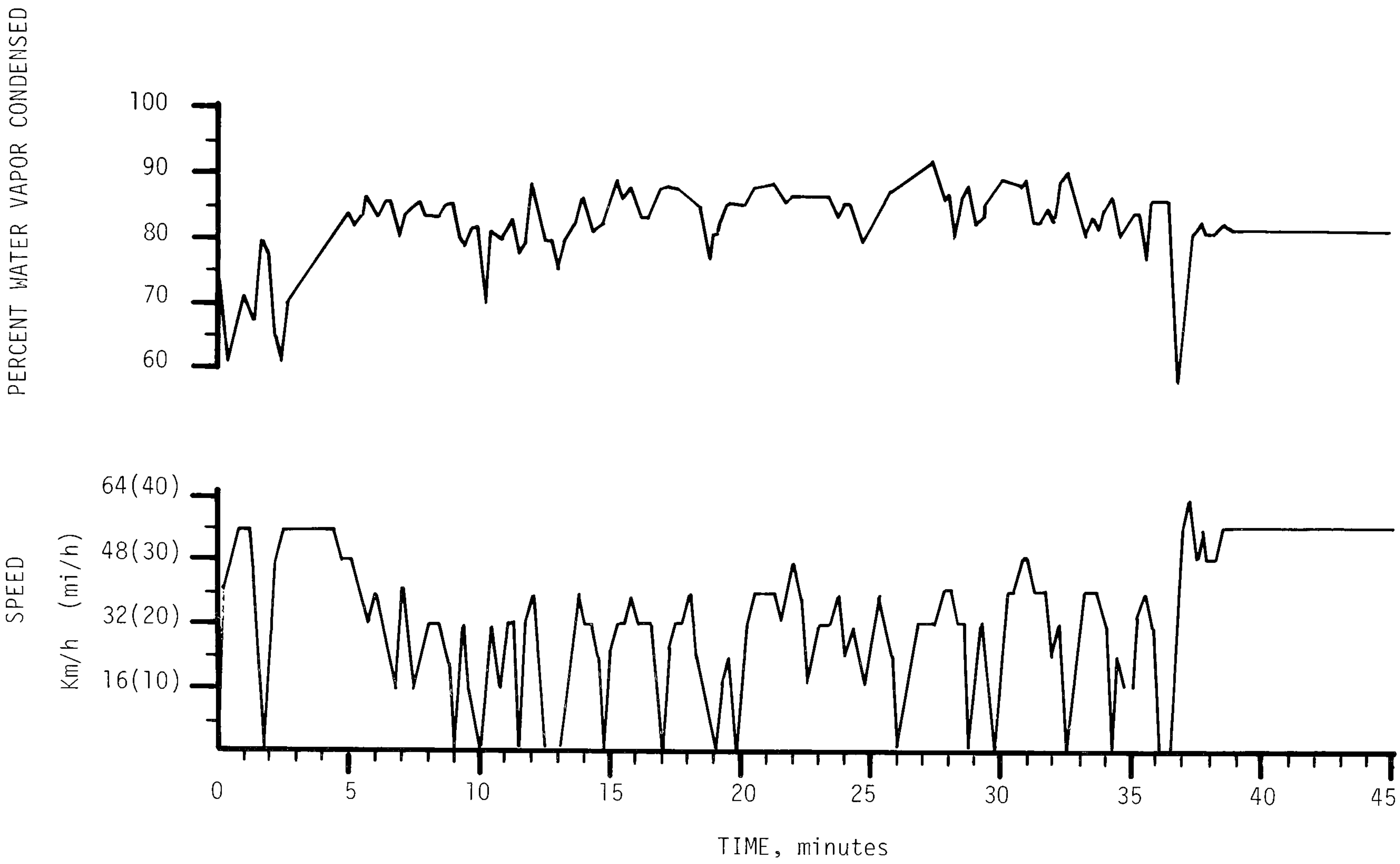


Figure 11. Cooler-condenser performance during a drive through town.

of the muffler. Extremely high back pressure readings on the 1968 Scout (Table 2) were probably caused by some tubes freezing shut. All other contractors reported insignificant back pressures with the addition of a cooler-condenser.

The back pressures attributable to the first winter's cooler-condensers are shown in Figure 12. Addition of the cooler-condenser to the Jimmy's exhaust system increased the back pressure by 4-1/2 times at 80 km/h (50 mi/h). However, except for high speeds, its back pressure was similar to that on the Nova which had the same displacement engine.

The first cooler-condenser on the Nova was the only one which did not substantially increase back pressure much above the normal exhaust system. However, twice during highway runs at temperatures below -29°C (-20°F) the cooler-condenser plugged with ice and caused the pressure relief cap on the end of the normal exhaust pipe to blow off. Apparently the small, thin tubes on this first condenser (modified radiator) froze shut.

The back pressure on the diesel was higher than for the gasoline powered vehicles because, for the same fuel consumption, the diesel engine has about three times the exhaust volume (200 percent excess air).

The effect of engine rpm for a given road load (constant at any one speed) is shown by the third and fourth gear curves (Figure 12). Use of the third gear at 64 km/h (40 mi/h) creates more exhaust gas than fourth gear, which results in more back pressure. A discussion of the diesel back pressure as related to chain turbulators in its cooler-condenser during the second winter is in Appendix A.

Because of greater temperature differences, the overall condensation performance as indicated in Figures 6, 7, and 8 should have increased as the temperatures decreased. But they did not. It was speculated that there was some internal ice film formation increasing the thermal resistance (reciprocal of the heat transfer coefficient). If so, this ice film would have reduced the flow passage (cross-section area) resulting in increasing back pressure with decreasing ambient temperature (more ice). The effects of this icing are demonstrated in Figure 13, which shows that at the lower ambient temperatures there is a significant increase in back pressure.

The icing effects did not appear to be accumulative over short periods because the stainless steel cooler-condenser on the Nova never plugged with ice. However, the vehicle was parked in a heated garage about once every 5 to 15 days. But most of the time the garage temperature at the cooler-condenser level never exceeded 0°C (32°F). Therefore, any ice should not have melted in the garage.

On the morning of February 12, 1976 the 1968 Carryall (4x2) was left idling at about -30°C (-22°F) for 30 minutes while a local television camera crew made a filmed newscast for the evening news. The vehicle was shut off without blowing out the liquid condensate which had accumulated at the bottom

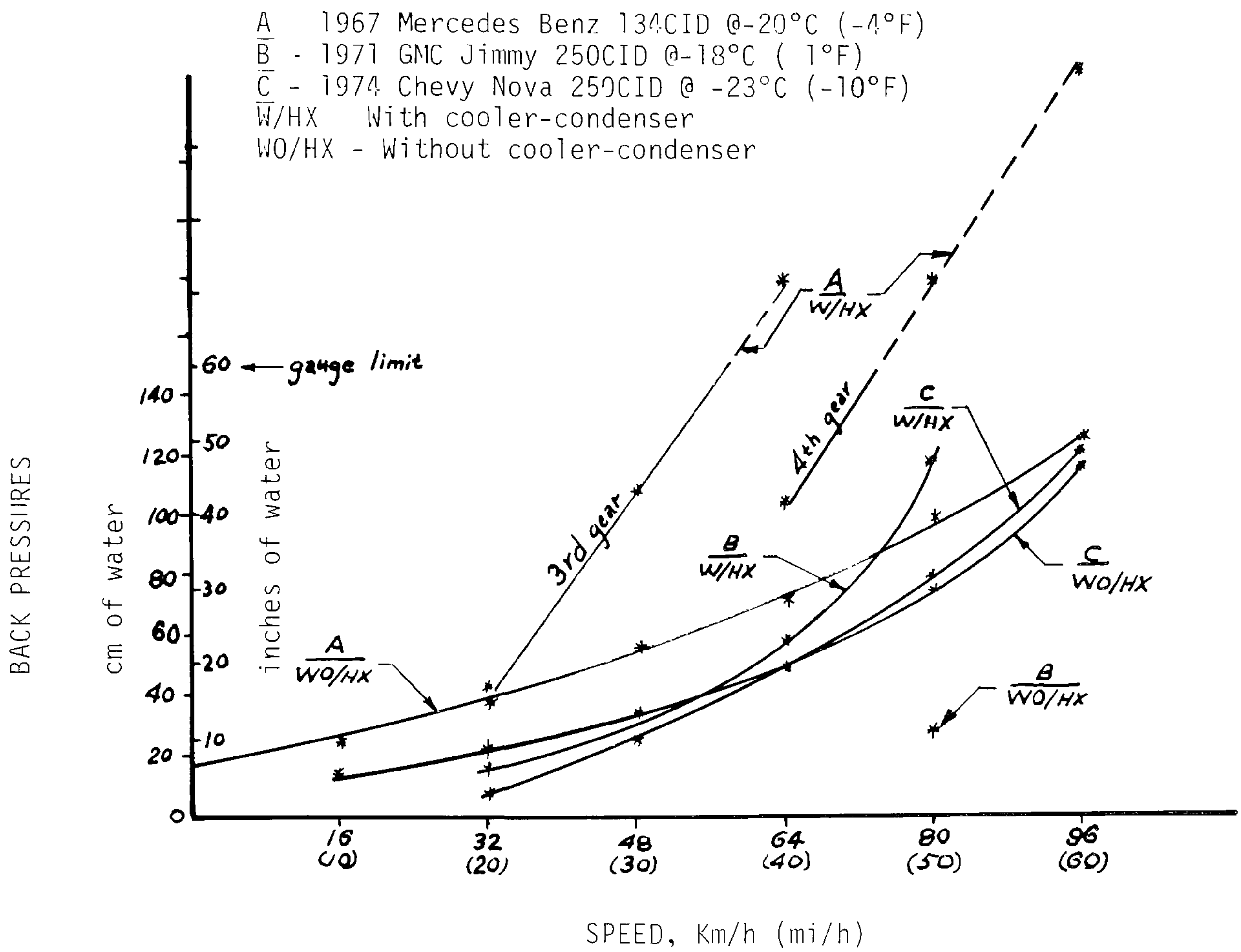


Figure 12. First winter cooler-condenser back pressures.

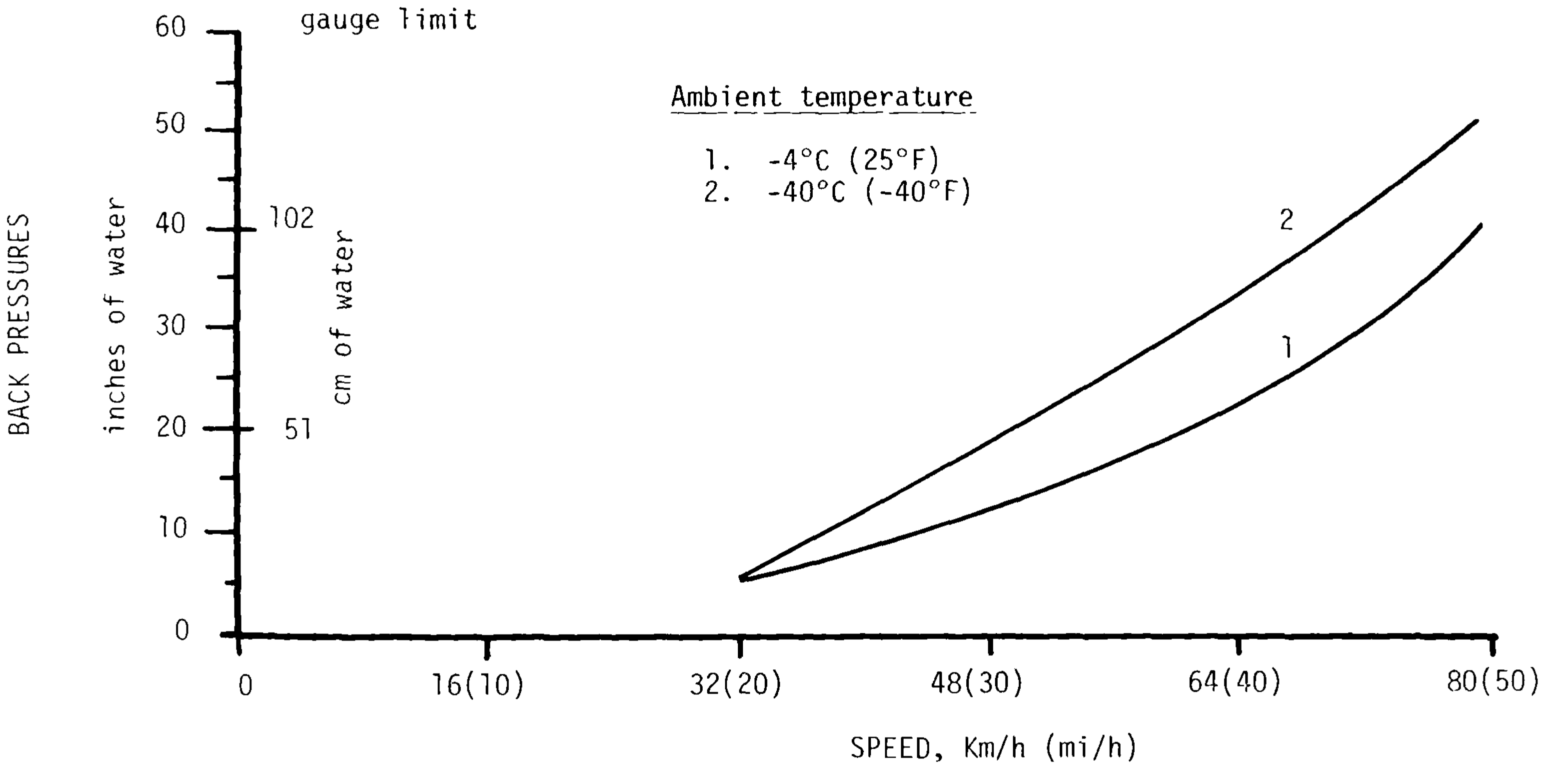


Figure 13. 1974 Chevy Nova second winter cooler-condenser back pressures.

of the cooler-condenser. Later attempts to use the vehicle were futile because the cooler-condenser system was plugged with ice. The back pressure was so great that the vehicle would only idle. It was therefore driven, at two miles an hour, to a heated garage where it thawed open after sitting inside over a weekend. Even with the garage thermostat set at about 20°C (68°F) it took three days to melt the ice. This situation was probably precipitated as a result of intentional plugging of the weep hole on the bottom header. The hole had been plugged because it was too large, allowing excessive exhaust gas leakage. There were no other ice plugging problems with the EMT cooler-condenser on this Carryall.

Back Pressure and Fuel Economy

Exhaust systems are generally designed for low back pressure because it is known that high back pressures rob power and decrease fuel economy. The cooler-condensers increase back pressures, as shown in Figures 12 and 13. During the second winter it was decided to quantify the back pressure effect on fuel economy for two vehicles.

Test runs were made in the Nova and Jimmy over a 32 kilometer (20 mile) level test section on the Richardson Highway, south of Fairbanks. Fuel economy was measured both with and without cooler-condensers in the exhaust system.

The back pressure with the cooler-condenser was from 20 to 150 percent more than without it. For example, back pressure on the Jimmy at 80 km/h (50 mi/h) was steady at 51 cm of water column (20 in. H_2O) with the cooler-condenser connected. Without the cooler-condenser it was 36 cm of water column (14 in. H_2O). For the Nova, at 80 km/h with the cooler-condenser the back pressure was 76 cm of water column (30 in. H_2O), and 31 cm water column (12 in. H_2O) without.

Calculations with data from the Kent-Moore fuel meter (readable to 0.001 gallon) showed there was only a 0.5 percent increase in fuel consumption with the cooler-condenser on the Nova, and 3.2 percent increase with the exhaust dilutor-ambient air heater on the Jimmy. The precision and accuracy of the instruments were each less than 4 percent. Therefore, this suggests that additional back pressure due to installation of cooler-condensers will probably not reduce fuel economy by a significant amount.

Passenger Compartment Carbon Monoxide Measurements

When installing a cooler-condenser, the final exhaust outlet was relocated to the front of the vehicle in some cases. This relocation might increase the risk of introducing more exhaust fumes into the passenger compartment, especially if the exhaust is directed out the drivers side, against an adjacent wall such as at a bank drive up window.

To quantify this risk, interior carbon monoxide (CO) levels were monitored periodically on test runs. Measurements were taken with an Ecolyzer, ambient carbon monoxide analyzer, model #2900. Values did not follow any pattern except at speeds above 32 km/h (20 mi/h). With the windows closed and heater running the CO level remained at or near zero parts per million (ppm) on the 1974 Chevy Nova, which was equipped with a front mount cooler-condenser and front exhaust outlet. At idle the reading ranged between 3 ppm and 30 ppm with an average of 22 ppm. For speeds up to 32 km/h (20 mi/h) the average was about 7.5 ppm.

The 1968 Chevy Carryall cab readings were higher because the two cooler-condensers mounted at the front of the vehicle were not leakproof and there were a number of holes in the floor boards through which exhaust gases entered. The exhaust outlet was directed at the drive shaft behind the front seat. At idle the CO readings ranged between 10 ppm and 40 ppm with an average of 18 ppm. At speeds up to 32 km/h (20 mi/h) the levels ranged between 5 ppm and 20 ppm with an average of 11 ppm.

The 1971 GMC Jimmy also had high passenger compartment CO levels with the exhaust dilutor-air heater mounted under the vehicle. One reason for this was that some of the distribution holes were directed towards the floor boards, giving exhaust gases more opportunity to enter the cab. For periods of idle lasting over two minutes, CO levels reached a maximum of 100 ppm. Other idle data gave values between 50 ppm and 10 ppm with an average of 33 ppm. Speeds of between 16 and 32 km/h (10 and 20 mi/h) had CO readings between 50 ppm and 15 ppm with an average of 26 ppm. Above 32 km/h (20 mi/h), the CO level dropped below 5 ppm.

In all of the above measurements, none of the vehicle's occupants were smoking. For comparison, one pipe smoker in a pickup truck raised the cab CO levels at different rates, depending on certain variables. For example, at idle, with the heater on the CO was 25 ppm. Turning the heater off increased the level to 45 ppm. Cruising at 56 km/h (35 mi/h) with the heater on, the CO was 5 ppm; however, shutting off the heater increased it to 15 ppm. A puff of smoke blown in the direction of the CO analyzer shot the CO level to over 60 ppm.

Overall Ambient Ice Fog Reduction

In discussing ice fog control techniques, the first question one asks is, "How effective is it or how much will the fog be reduced?"

Ice fog is generated by heating and power plants, cooling ponds, and motor vehicles. Ice fog from the tall stacks of power plants does not readily add to the ground level problem until the temperature drops to -40°C (-40°F) or less. Again, except for very low temperatures, cooling pond ice fog is mainly concentrated adjacent to and down wind of the ponds. There are only two cooling ponds of significance in the Fairbanks air shed. They are the Ft. Wainwright pond which serves the South power plant, and the Chena River which acts as a receiver for the cooling water from the Fairbanks Municipal Utilities System.

When considering on-the-road ice fog, there are two significant sources; low level home heating stacks and motor vehicles. At this point, we will consider only these latter two sources in calculating ice fog reduction.

From discussions with the fuel suppliers, the authors estimate that the amounts of heating oil and motor fuel consumed in the Fairbanks air shed are about equal; 50 percent each. The amount of water vapor created is roughly the same for each of these fuels. Therefore, if controls were applied to motor vehicles to reduce their water vapor emission by 80 percent, then the overall ground level water vapor emission would be reduced by one half of 80 percent, or 40 percent.

Let us consider two conditions, one with incipient ice fog and the other with ice fog already present.

As a first example, without ice fog control, assume conditions are such that incipient ice fog occurs at -32°C (-25°F). Incipient means 100 percent relative humidity @ -32°C. And also assume 90 percent of all water vapor in the affected area is a result of man's activities. Now, if ice fog control is applied to all motor vehicles, when would ice fog begin to form? From the psychrometric charts (Appendix E @ -32°C [-25°F]) the saturation (incipient ice fog) humidity ratio is 0.0002 gm of water per gm of dry air. Now if ice fog controls were to decrease the water vapor input by 40 percent, the humidity ratio would be reduced to 90 percent of (1 - 0.4) 0.002 = 0.001 which would not appear as ice fog (100 percent relative humidity) until temperatures reached -37°C (-35°F). Because the air is very stable there will still be isolated spots, such as at intersections, where ice fog (relative humidity >100%) will still appear. The above calculations assume complete phase equilibrium; which means that there is no supercooled water vapor.

When ice fog is present the air is saturated with water vapor. Therefore, any water vapor emission will immediately be converted into more ice fog. The Lambert-Beer Law can be used to better estimate the visibility effect of reducing the ground level ice fog input by 40 percent. This law of light absorption states that:

Light intensity after absorption = initial intensity x $10^{-\alpha bc}$, where α is the extinction coefficient which depends upon the ice fog particle morphology, b is the sight distance, and c is the concentration of ice particles. When it is desired to calculate the effect of a new concentration (c') on sight distance (b') for the same light intensity the relationship reduces to the follow ing form:

$$\alpha' b' c' = \alpha bc$$

When considering ice fog from only one source $\alpha' = \alpha$; then: $c'/c = b/b'$.

If control reduces ambient ice fog by 40 percent then c' = 0.6c. The effect upon the sight distance is b' = b/0.6 = 1.67b. Therefore, for this case the visibiltiy is increased by 67 percent. Based on all assumptions discussed above, this means that if visibility were 50 meters (150 ft.) before control it would then be 83 meters (250 ft.) after control. The visibility increase would be slightly greater than 67 percent because the extinction coefficient for motor vehicle created ice fog is higher than for ice fog from other sources. Exact values of this coefficient for the various types of ice fog are unknown.

The overall impact of automotive ice fog control would be a visibility increase of 70 percent or more in areas where motor vehicles create 50 percent or more of the ice fog. The picture will not always be as favorable as shown because of the many other sources of ice fog. However, the motor vehicle ice fog control devices would greatly reduce the roadway ice fog and prevent dense ice fog from obscuring busy intersections.

If the water vapor is no longer put into the air, where does it end up? The ice fog control device condenses it to liquid water which readily freezes and if not retained by the vehicle undercarriage, it falls onto the road. The road icing effect will be discussed in Section 7.

SECTION 6

FUTURE APPLICATIONS

SELECTION OF HEAT TRANSFER MEDIUM

Both the exhaust dilutor-air heater and the cooler-condensers are effective mobile source ice fog control devices. But because of its intentional exhaust dispersion beneath the vehicle's floorboards, the exhaust dilutor-air heater should not be used under any passenger compartment.

The cooler-condenser is the only remaining safe, effective device. To keep the physical size within reason requires forced convection. However, ice films may develop which cause poor performance and increased back pressure. Weep holes in the manifolds are not effective in reducing ice films in condenser tubes. There are three possible solutions. One is to add control louvers, as with the 1968 Scout, to block the cold air. Or the exhaust flow passages can be made so large that they will hold a winter's ice film buildup. It would then be necessary to oversize the surface to compensate for increased heat transfer resistance. The last solution would be to use cold antifreeze for the coolant as was done on the 1968 Jeep, where a baffled radiator was the cooler-condenser. Freezing problems would be less since the flow area for the exhaust gas would be 5 to 10 times that of a nominal 5 cm (2 in.) tail pipe. It would take much longer for ice to bridge across the fins. Also, additional freeze protection could be accomplished by incorporating a temperature controller which would block off air flow to the normal radiator causing the antifreeze to warm up enough to melt ice in the cooler-condenser.

The other advantages of the baffled radiator cooler-condenser are: low space requirement because of the extended surface, low pressure drop because of large flow area, and low cost because of mass production of radiators. If automotive radiator solders, which soften at about 200°C (400°F) can withstand the high exhaust temperatures on one side and cold antifreeze on the other, then economical cooler-condensers could be fabricated by adding baffles to an auxiliary radiator as was done on the 1968 Jeep. The automobile's regular radiator would cool the antifreeze before being pumped through the cooler-condenser (baffled radiator). The disadvantage of this system is increased vulnerability to the engine cooling system. The additional hoses and connections to the antifreeze system increase the chance of leaks or total loss of coolant.

MOUNTING LOCATIONS

When considering air to exhaust gas cooler-condensers the first question that comes to mind is where to put them. To obtain the best advantages of forced convection they should be mounted in front or behind the vehicle's radiator. Many American-made vehicles have a removable spacer on the radiator fan hub. Moving the fan back allows mounting room for a cooler-condenser.

The EMT cooler-condenser on the 1968 Chevy Carryall (4x2) was mounted in front of the radiator. To check for possible hot weather problems a long 2 km (1.2 mi) run up a 12 percent grade during a warm 24°C (75°F) day did not cause the engine to overheat even with hot exhaust flowing through the EMT cooler-condenser. Cooler-condensers could also be designed to fit into the space reserved for the automotive air conditioner condenser, since air conditioners are unnecessary in arctic regions.

CORROSION

The cooler-condenser condensate is corrosive because it contains carbonic halogenic, sulfurous(ic), and nitrous(ic) acids. One contractor reported the condensate pH to be consistently between 3 and 5.5 which is highly acidic. He recommended that anodized aluminum be considered for these cooler-condensers (1). Because of the difficulty in maintaining the integrity of the protective aluminum oxide film during fabrication, the authors feel that anodized aluminum would quickly corrode. Regular or mild carbon steels would also not be expected to last long when subjected to this acidic condensate.

A recent material engineering symposium addressed the corrosion problems in flue gas scrubbers (14). The chemical quality of the scrubbing liquid would be similar to the cooler-condenser condensate. The following recommendations were made: use stainless steels such as Carpenter 20 or Uddeholm 904l when the pH is below 4 and no halides are present. This would mean the use of unleaded gasoline, because leaded gasolines contain bromine compounds to act as lead scavengers. For condensate from leaded fuel, Hastelloy "C" or Inconel 625 should be used. These alloys are very expensive, costing more then $1 per pound when fabricated into tubing.

An exhaust system manufacturer recommended the use of a special "muffler steel" known as type 409 Stainless. Since its cost is much less than the other stainless steels it was decided to use it in the second winter's cooler-condensers. Type 409 is a 12 percent chromium ferritic steel used in the exhaust systems of some of the 1975-76 automobiles. It costs less than $1.50 per kg (70¢ per pound).

The tubes for the second cooler-condensers on the Chevy Nova and Mercedes diesel were fabricated from type 409 Stainless. So far there has been no evidence of corrosion. The EMT and galvanized flexhose have shown considerable corrosive attack in the same time period.

Old automobile radiators have been used as the heat exchanger for the antifreeze coolant cooler-condensers. There has been no corrosion evaluation but it is expected that the copper alloys would stand up as well as the stainless steel.

COSTS

The installed costs of the ice fog control devices varied from about $100 for the flexhose exhaust dilutor-air heater to about $850 for the stainless steel cooler-condenser. It should be remembered that these prototype devices were one-of-a-kind fabrication. If they were mass produced the installed cost would probably be 1/3 to 1/2 the above. As an example of the cost reduction by mass production, consider the $90 automobile radiator (heat exchanger). If it were to be fabricated as a one-of-a kind, the shop cost would exceed $500.

The lowest cost, most readily available, solution to automotive ice fog may be to use automobile radiators (as with the 1968 Jeep) as the cooler-condensers. This is because they are mass produced and contain more surface per dollar than any other heat exchanger. But they should be evaluated with higher temperature exhausts to see if solder melting and corrosion are problems.

As mentioned in Section 6 the flexhose exhaust dilutor-air heater should not be considered for use under passenger vehicles.

RECOMMENDATIONS

All the control devices worked to some extent on all the vehicles. There were no insurmountable problems, although condensate freezing could be a real problem in improperly designed cooler-condensers. What is now needed is a demonstration-evaluation on fleet vehicles that are routinely operated in the dense ice fog areas. Some vehicles should try antifreeze as the coolant, while others could use the forced convection air cooled cooler-condensers. The fleet testing would further demonstrate the practicability of automotive ice fog control to the general public and would provide enough information to select the best coolant and to decide if automobile ice fog control regulations are desirable.

SECTION 7

ENVIRONMENTAL CONSIDERATIONS

ADDITIONAL ICE ON THE ROAD

The accumulation of ice on public roads and at intersections due to deposits made by vehicles equipped with cooler-condensers has given rise to this question, "What increased dangers will be encountered by the additional ice in relation to the hazards of poor visibility created by ice fog?" The overall percentage of ice added to the highway has been calculated to be about 20 percent more than what Mother Nature deposits over the four month winter period (Appendix D). Assuming the condensate leaving the cooler-condenser hits the road surface in the form of water, then freezes, the total accumulated ice is approximately 1.4 cm (0.54 in.) from November through February. This estimated condensate accumulation amounts to about 1/3 cm (1/7 in.) per month. Mixed with accumulated hoar frost and snowfall it would be difficult to identify and measure. Some of the smaller vehicles with large exhaust systems, which act as cooler-condensers, presently deposit some condensate on the road during extreme (-40°C) weather. This deposit is presently concentrated at left turn lanes.

The Alaska Department of Highways indicates that intersections are sanded or plowed according to need. No other criteria are considered. Since sanding is done only at intersections, bus stops, and curves, very little economic impact should be felt by the state Department of Highways with the incorporation of cooler-condensers on automobiles. Danger from the additional ice is minimal in comparison to the hazards of low visibility driving in ice fog. But the effects may be conflicting. Better visibility means many drivers will increase speed, which may be more dangerous on the increased ice.

To further qualify the geometry of this fallen condensate, a piece of sheet metal plate 1 x 1.3 m (3 ft x 4 ft) was mounted under the 1968 Chevy Carryall (4x2) to simulate a roadway. Its purpose was to catch condensation leaving the cooler-condenser to determine the shape and pattern of the condensate hitting the road, and whether or not the condensate would freeze before reaching the road.

First tests were done using a 8 cm x 45 cm (3x18 in.) piece of plastic pipe coalescer mounted against the vehicle frame. Condensate leaving the coalescer when dropped a distance of 5 cm (2 in.) ran onto the metal plate forming a sheet of ice in an alluvial fan pattern approximately 3.8 cm (1.5 in.) thick and 25 cm (10 in.) wide. The coalescer was removed to provide a long drip (distance) resident time. It was thought that the latent heat of freezing would be dissipated at ambient temperatures below -23°C (-10°F) before striking the metal sheet. The condensate fell in a liquid state forming an ice sheet as before. The vehicle was driven at approximately 32 km/h (20 mi/h) with many stops and turns to simulate city driving. 1.9 liters (0.5 gallons) of gasoline were consumed. This should have yielded approximately 1.8 liters (0.4 gallons) of condensate, 2/3 of which was collected on the metal plate.

Next, a chain followed by a wire screen was placed between the outlet and the metal plate. A long icicle formed betewen these intermediate devices and led to a pool which froze as a sheet of ice on the metal plate.

The outlet was then placed at a 30° angle with respect to the vehicle frame and 0.6 cm (1/4 in) above the drive shaft. Ice formed around the spinning shaft much like the quills on a porcupine. Ice formed a rough surface on the metal sheet in a scattered pattern radial with respect to the outlet above the drive shaft. Some ice formation accumulated on the frame and the undercarriage as the condensate spun away from the drive shaft.

Experience with the above techniques indicates that most of the condensate would fall upon the road surface as droplets and freeze into rime ice. During extreme cold the condensate would probably freeze before reaching the road surface. When the ambient temperatures are only slighty below freezing, the condensate would freeze to the road as clear ice. However, where there is heavy traffic, tires would erode the forming ice through attrition and cause some of it to be swept to the roadside. Also, some of the condensate would freeze before reaching the road surface. It would not then easily adhere, but would be partly swept aside by traffic.

REDUCTION OF OTHER AIR POLLUTANTS

Passing gas through a water spray, known as gas scrubbing, is one of the oldest techniques for cleaning gases. The condensing section of a cooler-condenser is an excellent scrubber because some of the small submicron particles that might not be caught in a water spray will act as condensation nuclei and thereby be removed. The condensing section should theoretically remove the majority of the exhaust particulates such as lead compounds and soot. Since carbon monoxide is not appreciably soluble in water, it is not expected to be removed in the condensation process. There should be partial removal of the water soluble toxic gases such as halogenic acids, sulfur oxides and nitrogen oxides.

Carbon dioxide is also absorbed in the condensate to form carbonic acid. The fact that these acids are present in the condensate is indicated by its low pH which ranges from 3.5 to 5.5.

A test was run on the 1974 Chevy Nova cooler-condenser to determine what fraction of the exhaust lead compounds and sulfur oxides were actually removed with the condensate. The vehicle was allowed to idle at an ambient temperature of -36°C (-32°F) while its condensate was collected. The gasoline consumption was measured with a 0.1 gallon (0.4 liter) burette and the resultant condensate with a graduated cylinder. The test was divided into three consecutive runs so that the amounts of collected condensate could be used to verify steady state conditions. The calculated percent water vapor condensed varied from 85 to 88.

The sulfur oxides come from the combustion of sulfur compounds in the gasoline. They are measured as sulfates in the condensate. One contractor performed such analysis and reported sulfate levels from 8 mg/l (milligram per liter) to 134 mg/l (1).

The sulfur removal efficiency for the run on the Nova was calculated by a sulfur balance which says that all the sulfur in the fuel must show up as sulfur oxides in the exhaust. The test was run with the vehicle idling at -36°C (-32°F). Exactly 1.1 liter (0.3 gallons) of gasoline was burned. The fuel supplier said it contained 0.2 percent sulfur. Assuming complete combustion the sulfur oxide emission would be 170 mg as sulfur. The collected condensate amounted to 0.695 liter which had a sulfur compound concentration of 32 mg/l as sulfate; 7.4 mg as sulfur. The analysis was the Barium Chloride Turbidimeter method. Therefore, four percent of the exhaust sulfur compounds were captured in the condensate. Extrapolating for the range of reported sulfate concentrations the cooler-condenser can be expected to capture from one to 20 percent of the sulfur oxides.

With the introduction of catalytic converters on the newer vehicles some of the sulfur dioxide is further oxidized to sulfur trioxide which reacts with exhaust moisture to form a sulfuric acid mist. The trioxide and sulfuric acid mist have a much higher affinity for water than does sulfur dioxide. Therefore, on converter equipped vehicles, cooler-condensers would be expected to remove a larger fraction of the sulfur oxides.

Atomic absorption analysis for lead in the Nova condensate yielded 24 mg/l. But for condensate from the 1968 Chevy Carryall values of 54 and 190 mg/l were obtained. The fuel supplier indicated that the gasoline contained 1.4 ± 0.28g tetraethyl lead (TEL) per gallon. Since TEL is 64.1 percent lead (Pb), the Pb in the consumed gasoline (0.3 gallon) was 1.4 (0.3) 0.641 (1000) = 269 mg. At 24 mg/l the Pb in the 695 ml of Nova condensate amounted to 24 (0.695) = 17 mg. Therefore, the Pb removed in the condensate was 6 percent of that in the gasoline. That is much lower than expected. Corresponding removals for the cooler-condenser on the Carryall were 14 and 49 percent for the 54 and 190 mg Pb/l condensates, respectively. Record checks indicated that the Nova had not been fueled with lead free gasoline. Since its exhaust system and cooler-condenser had not been in service as long as those on the Carryall, they may have still been accumulating an internal coating of leaded combustion products. This could explain the discrepancy between the lead concentrations in the different condensates.

A simple nitrogen balance to determine the efficiency of nitrogen oxide removal by the cooler-condenser is not possible because an undetermined fraction of the atmospheric nitrogen is oxidized to nitric oxide. The solubility of nitric oxide (NO) is about 1/2000 that of sulfur dioxide. Therefore, the cooler-condenser removal efficiency for nitrogen oxides is expected to be insignificant.

Hydrogen halide (halogenic acids -HX) appears in the exhaust because halides are added to gasolines as lead scavengers. Their removal with condensate is expected to be relatively high since HX are from two to ten times more soluble than sulfur dioxide.

Other non-water soluble gases such as the lighter hydrocarbons (methane through propane) are not removed by cooler-condensers.

CONDENSATE QUALITY

As mentioned in the section on Corrosion, the condensate is very corrosive because it contains combustion products such as carbonic, hydrohalic, sulfuric, and nitric acids. Its pH varies from 3.5 to 5.5. It also contains lead (leaded gasoline) compounds and soot. When released into the environment its pH will slowly approach neutral as the carbonic acid decomposes into carbon dioxide and water.

The condensate total solids content varies from 800 to 3900 mg/l. The larger number represents some of the cooler-condenser corrosion products (rust). About 60 percent of the solids are volatile (burn off at 600°C), which indicates that bicarbonates and soot may be the major solids in the condensate.

Without any ice fog control most of these combustion products are captured in the ice fog particles which eventually settle on the Chena River flood plain. During breakup these compounds are released and captured by the soil or its vegetative mantle or they flow into the local streams. This acidic ice fog melt is probably well buffered before it reaches any water course.

With ice fog control the frozen acidic condensate will be removed along with snow cleared from local roads. This snow is hauled to snow dumps -- some on the banks of the Chena River. Upon melting it will drain directly into the river. This low pH melt will probably not cause any pH drop in the river because there is enough alkalinity available to act as a buffer. Also the sulfur, nitrogen, and halide load to the river would show a short term increase during snow dump melt. The increase of lead and soot would probably be less since considerable lead and soot already end up in the snow dump.

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APPENDIX A

DETAILED DESCRIPTION OF ICE FOG CONTROL DEVICE INSTALLATIONS

COILED METAL FLEXHOSE COOLER-CONDENSER

Research on solutions to the automotive ice fog problem was started at the AERS in 1972. During that winter, a cooler-condenser heat exchanger was mounted underneath the carriage of a 1968 Chevy 1/2 ton Carryall with a 250 CID 6 cylinder engine. This cooler-condenser consisted of 150 feet of 2 in. spiral-wound, galvanized, flexible exhaust hose. The hose hung from the floorboards in four loops. The loops were hung over the propshaft under the frame channels and behind the transmission. The condenser was supported by 1-1/2 in. x 1-1/2 in. angle iron rests which hung from the floorboards by 3/8 in. threaded rods. The muffler was removed and the flexhose connected directly to the exhaust manifold pipe. The layout is shown in Figure A-1.

In this first attempt, the flexhose was chosen because it was easy to install. It was thought that since it would not hold water, it would not freeze shut causing excessive engine back pressure.

The installation time was three man-days.

The flexhose appeared to be condensing out all the water vapor at ambients of -12°C (10°F) and colder. Operation of the vehicle at below freezing temperatures caused massive icicle formations which clung to the flexhose, breaking off during travel over rough spots, such as crossing railroad tracks. The outlet temperatures were so low that the condenser became plugged with frost. There was no noticeable increase in back pressure since the connecting joints, every 50 feet, and the flexhose itself were not gas tight. The following data were obtained at -12°C (10°F) ambient:

SPEED	40 MPH	50 MPH
Inlet temperature °F	1100	1200
Outlet temperature °F	10	10
Back pressure inches of water	12	27

The exhaust noise at idle measured 70 db at 4 feet on the A scale of a Scott instrument, Lab Model 451 ANSI type S3a sound intensity meter. A 1968 pickup with the same 250 CID engine, but conventional muffler gave approximately the same noise level.

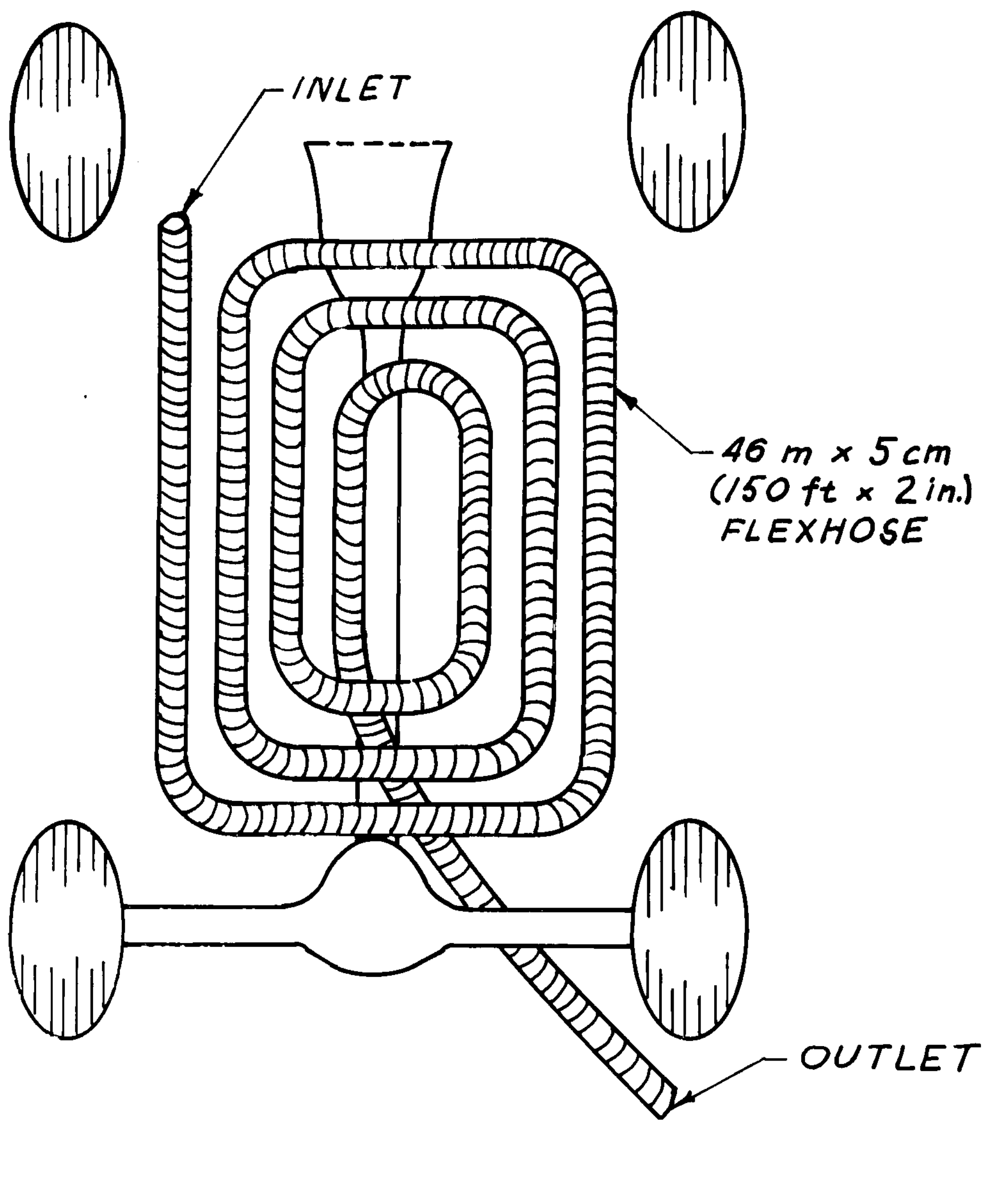

Figure A-1. Coiled metal flexhose cooler-condenser on 1968 Chevrolet Carryall (4x2).

The exhaust back pressure was about the same as that with a conventional muffler. For comparison, a 1974 Chevy Nova with the same displacement engine has a back pressure of 28 in. water at 50 mph.

COMBINATION COIL/TUBE COOLER-CONDENSER

In the spring of 1975, corrosion had weakened the coiled flexhose causing brittle failure at stress points. With the idea of trying out another heat exchanger, one was built from 1 cm (1/2 in.) electrical metal tubing (EMT) welded to two 5 cm (2 in.) header pipes (Figure A-2). Chains were inserted inside each tube to provide gas tubulation. It was mounted between the radiator and cooling fan. The fan hub spacer was removed to provide clearance. The outlet was located over the right side steering knuckle.

The EMT condenser did not provide enough surface area to adequately cool the exhaust gases. In November of 1975, two conventional steel mufflers were mounted in series in the exhaust pipe between the engine and the EMT condenser. Work was done by a local muffler shop. The mufflers increased the surface area to 1.1 square meters (12 square feet).

Although more surface area was provided, mounting of the mufflers under the vehicle prevented maximum air flow around them. The EMT outlet temperature was too high to allow condensation at ambient temperatures of -21°C (-6°F).

Another cooler-condenser was fabricated using 5 cm (2 in.) diameter flexhose mounted in a switch-back configuration in front of the radiator; adding one square meter (12 square feet) between the mufflers and the EMT. Also, the outlet was extended 2 meters (7 ft.) to the center of the vehicle. The total amount of surface area increased to 2.2 square meters (24 square ft.). An average outlet temperature at 64 km/h (40 mi/h) was 12°C (53°F) condensing out 85 percent of the water vapor at an ambient air temperature of -21°C (-6°F).

BRAZED RADIATOR COOLER-CONDENSER

The AERS automotive ice fog research effort was formally initiated during the winter of 1974-75. In that program, the first ice fog cooler-condenser was installed on a 1/2 ton GMC Jimmy with a 250 CID 6 cylinder engine. The first cooler-condenser was a radiator from a small foreign vehicle. Its overall dimensions were 13-1/2 in. x 12-1/2 in. x 4 in. Its solder joints were brazed to prevent joint melting. The solder melts at about 232°C (450°F) while the brass brazing rod would withstand exhaust temperatures to 454°C (850°F). The radiator was connected in front of the muffler where exhaust temperatures of 426°C (800°F) were expected. For protection, it was mounted between the drive shaft and the frame because the vehicle was used off the road and for snow plowing. The cooler-condenser performed satisfactorily, but had many attendant problems.

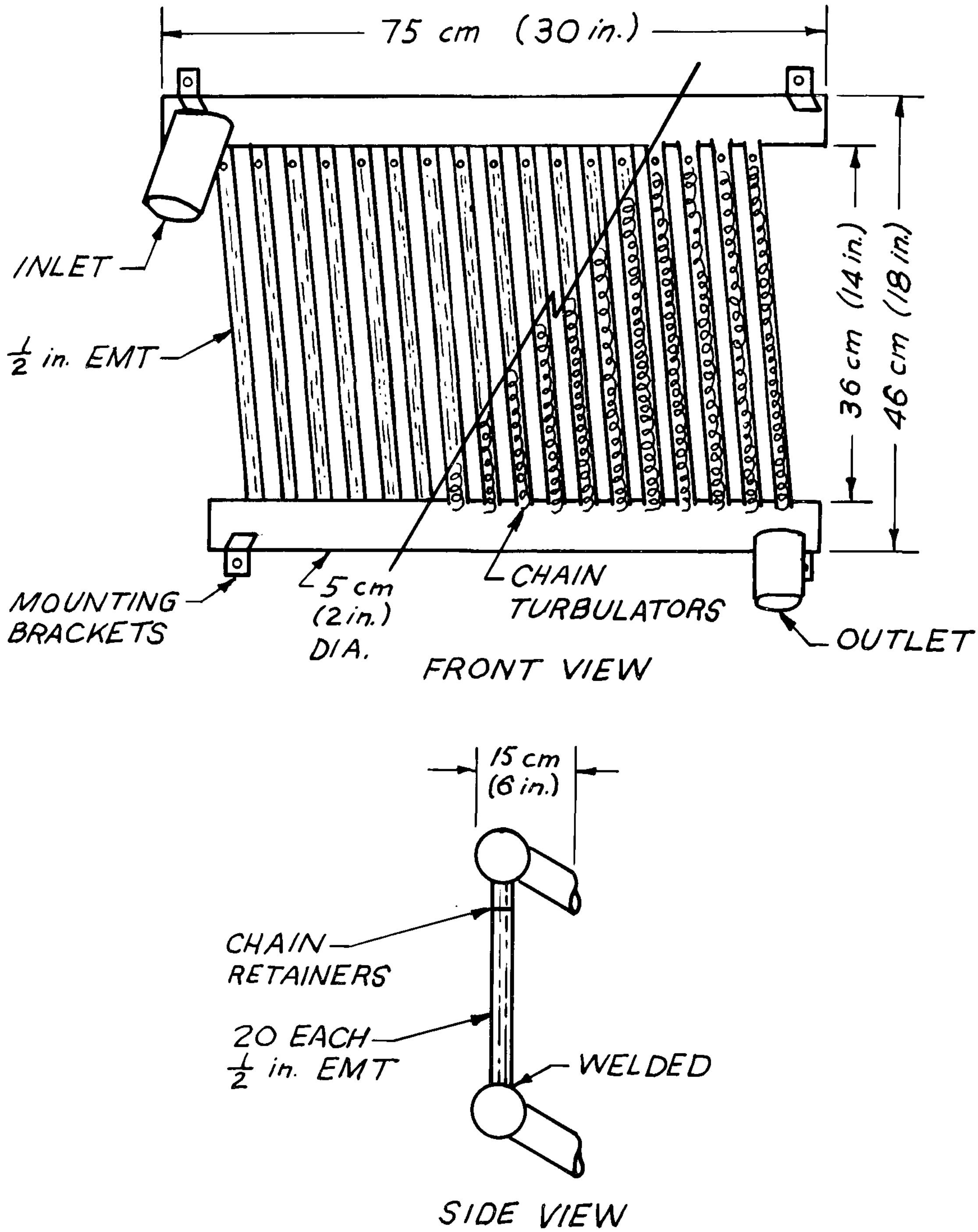

Figure A-2. EMT cooler-condenser on 1968 Chevrolet Carryall (4 x 2)

At an ambient air temperature of 2°C (35°F), the cooler-condenser dropped the exhaust from 321°C (610°F) to 32°C (90°F) at idle and from 482°C (900°F) to 54°C (130°F) at 64 km/h (40 mi/h). Extrapolating to an ambient temperature of -32°C (-25°F), it was estimated that the condenser exhaust temperature would be 2°C (35°F) at idle and 21°C (70°F) at 64 km/h (40 mi/h). The corresponding vapor (ice fog) removal would therefore be 95 percent and 82 percent.

It appeared that the condenser was going to be a success except that the high temperatures reached, 648°C (1200°F) at 88 km/h (55mi/h), caused the brazed header joints to melt open. These leaks resulted in excessive exhaust noise and high temperatures on the floorboards, so the cooler-condenser was removed.

FREE CONVECTION/FINNED TUBE COOLER-CONDENSER

Next, a heat exchanger was fabricated out of 1 cm (1/2 in.) electrical metallic tubing (EMT). The condenser had 10 tubes 40 in. long with 9 pieces of 1-1/2 in. x 3 in. slotted angle iron welded across the tubes to extend the surface area. The inlet and outlet manifolds were 5 cm (2 in.) exhaust tubing. This condenser was assembled and installed in about 1-1/2 man-days. It is shown in Figure A-3.

It did not provide enough surface area so one-half of a Young Radiator Company mobile oil cooler (MOC #6) was mounted in series with the EMT condenser. To fit, it was cut in half, parallel with the finned tubes; the internal tube turbulators were removed. Prior experience with the turbulators indicated they caused too high a back pressure. Overall dimensions of the 1/2 MOC #6 were 14-1/2 in. wide x 29 in. long x 1-1/2 in. thick, as shown in Figure A-4. The EMT condenser was hung in place of the normal muffler--between the frame and the drive shaft on the driver's side. The 1/2 MOC was mounted on the passenger side between the frame and drive shaft. The 1/2 MOC plus EMT cooler-condenser system condensed out 55 percent of the water vapor at idle. The total material cost is estimated at $200. The excessively large cooler-condenser surface area under this vehicle was not very effective because it was mounted high up between the frame channels. This shielded location prevented cold ambient air from effectively removing the exhaust heat.

EXHAUST DILUTOR-AMBIENT AIR HEATER COOLER-CONDENSER

During the winter of 1975-76 a decision was made to try a different method to solve the problem of ice fog control on the 1971 GMC Jimmy (4x4). It must be realized that ice fog, by definition, occurs only when the air is over-saturated with water vapor. For example, from the psychrometric chart, Fig. E-1 assumes an air temperature of -32°C (-25°F). At this temperature air can hold up to 0.0002 g of water per g of dry air. The 0.0002 g/g represents 100 percent relative humidity. Any additional water vapor input will form ice fog. But if the air is warmed to say -18°C (0°F) with the same water vapor content, the relative humidity will be reduced to 23 percent.

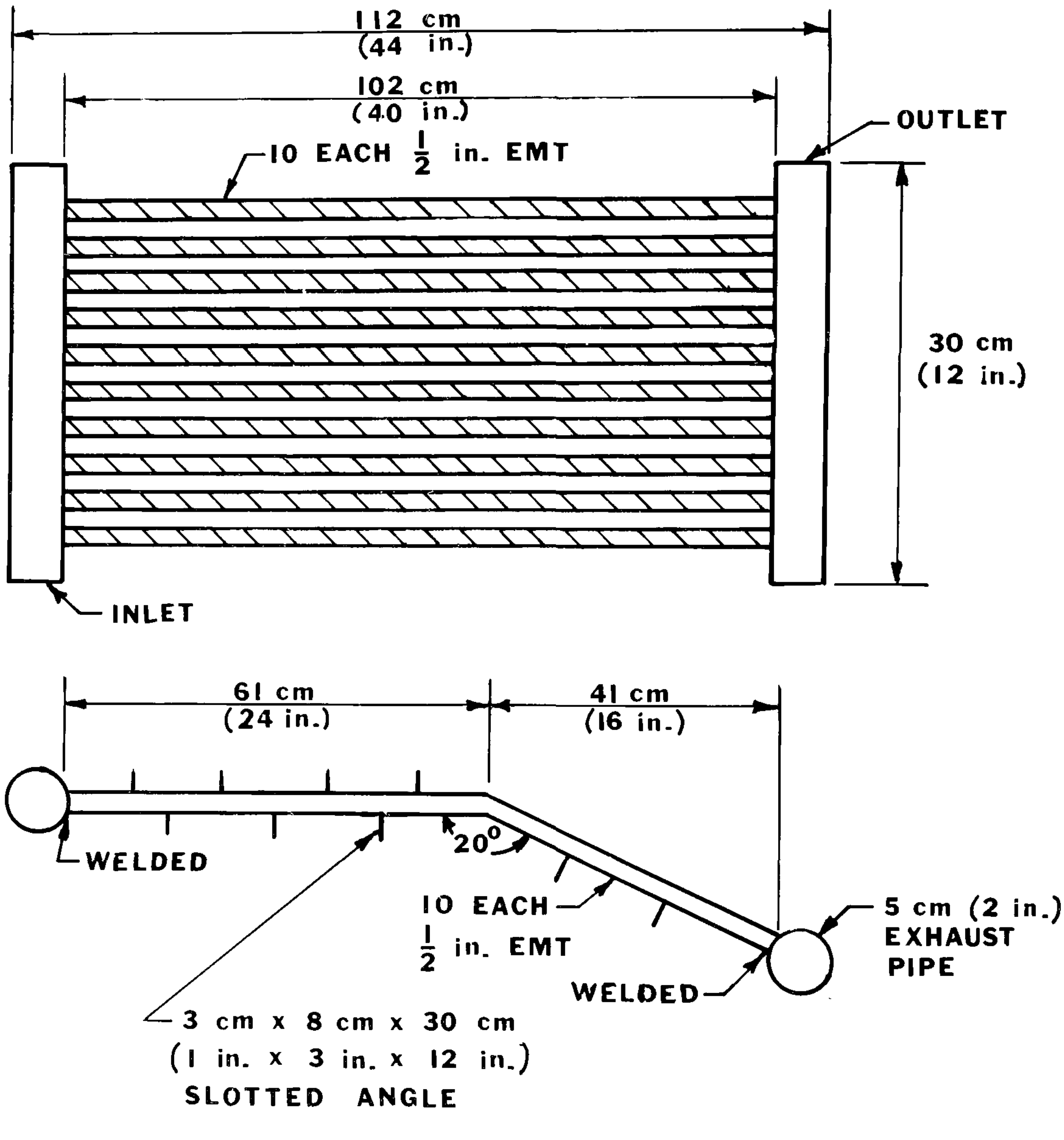

Figure A-3. First winter EMT cooler-condenser on 1971 GMC Jimmy.

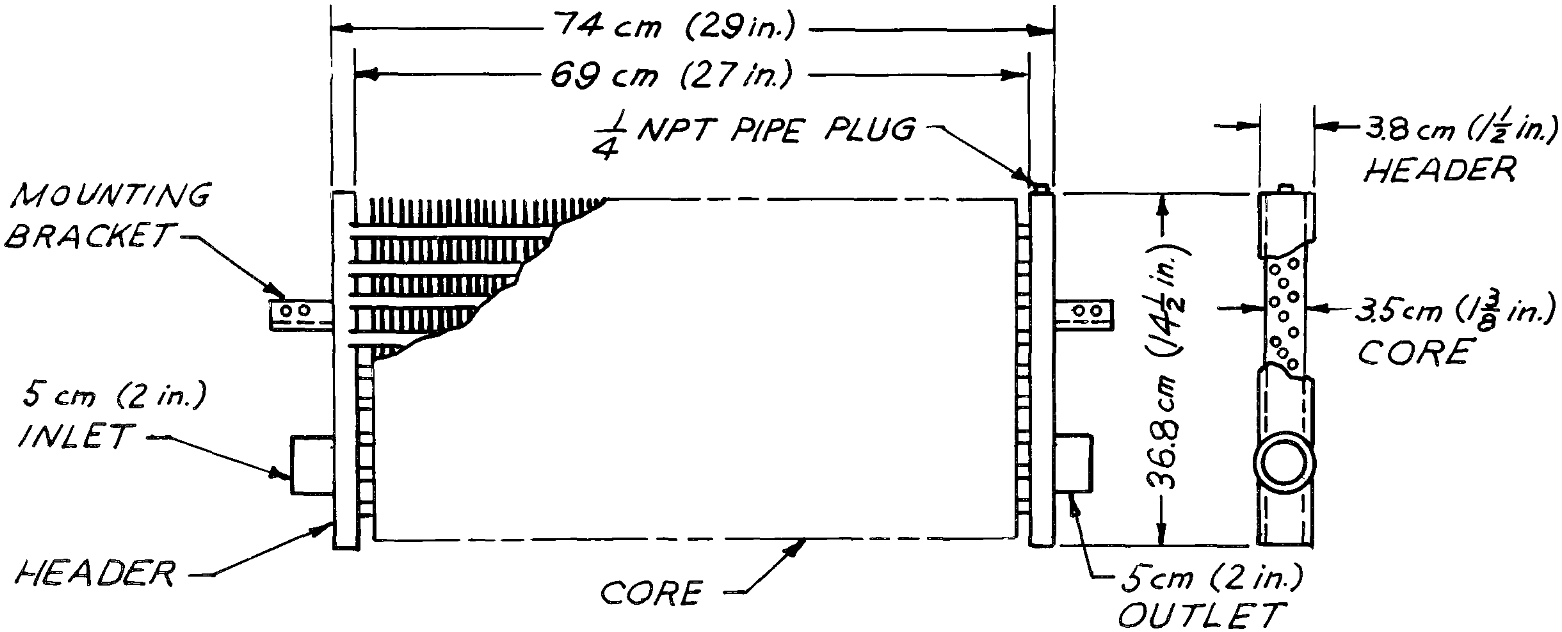

Figure A-4. Free convection/finned tube (MOC-6) cooler-condenser on 1971 GMC JIMMY.

At -18°C (0°F) the air is capable of holding up to 0.0008 g of water per g of dry air before ice fog will form. Therefore, by taking 1.0002 g of saturated air at -32°C (-25°F) and warming it to -18°C (0°F), it could accept 0.0006 g of water vapor before forming any fog. However some ice fog will form as the air cools from -18°C. This principle was tried on the 1971 Jimmy during the winter of 1975-76.

Eight meters (26 ft.) of perforated spiral wound flexhose were connected to the tail pipe. The perforated holes, four per foot, were 0.6 cm (1/4 in.) in diameter, approximately 15 cm (5 in.) apart. See Figure A-5. The hose was wired to the frame channels, behind the transfer case and under the rear bumper. The setup was quite effective in reducing visible ice fog by almost one-half.

The hose was wired below the rear bumper so that it could easily be slipped on or off the tail pipe for comparison purposes. Forcing the exhaust to flow out the 0.6 cm (1/4 in.) holes was accomplished by partially plugging the hose end with a paper towel; then condensate froze shut the last one meter (3 ft.) of hose. During operation, exhaust heat would be transferred to the ambient air through the metal hose walls. Simultaneously moist exhaust would be dispersed into this warm air because it could now accept more water vapor before becoming saturated and showing visible fog behind the vehicle. Also during extreme cold -34°C (-30°F) or less, some of the exhaust water vapor condensed out and dripped from the metal hose. Several ice stalagmites 10 to 15 cm (4 to 6 in.) high formed on the ground under the perforated hose when the vehicle was left idling for 15 minutes at -40°C (-40°F). In one winter's operation the hose rusted through at the tail pipe connection. Accurate costs for the perforated hose could not be determined since this hose was salvaged from the initial coils under the 1968 Carryall. The estimated new replacement cost with stainless steel spiral wound flexhose is about $75.

The advantage of this system is the low cost. One disadvantage is the danger of CO poisoning. Because most of the exhaust is released under the floorboards, some of it leaks through. Therefore, this system is not recommended when occupants could ride in any contaminated airspace. However, this system would be quite satisfactory under the open cargo bed of a truck. The other disadvantage is that this system is not as effective as cooler-condensers in limiting exhaust water vapor.

BRAZED RADIATOR

Early success with using the radiator as an exhaust gas cooler-condenser on the Jimmy indicated that the initial solution to the problem of automotive ice fog may be close at hand. It was decided to install a small radiator (cooler-condenser) with brazed joints on a 1974 Chevrolet Nova sedan with a 250 CID 6 cylinder engine. The cooler-condenser was mounted in front of the normal radiator behind the grill so the ambient air would flow over the cooling fins. For convenience it was mounted off center on the driver's side. The exhaust pipe was taped just in front of the muffler and 4.4 cm (1-3/4 in.) flexhose was used to convey the exhaust to the cooler-condenser. Its

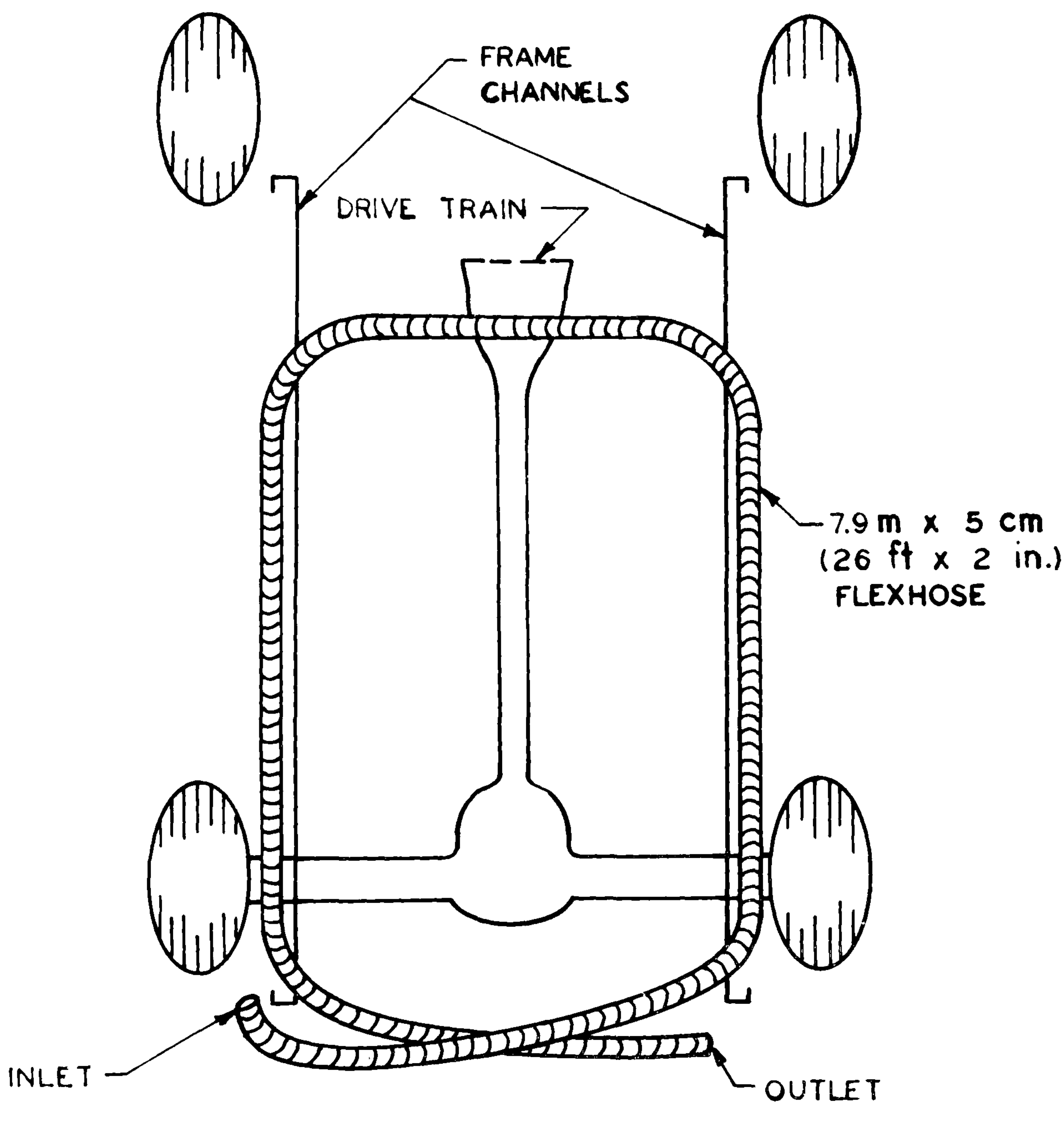

Figure A-5. Exhaust dilutor ambient air heater on 1971 GMC Jimmy.

overall dimensions were 39.4 cm (15-1/2 in.) wide x 43 cm (17 in.) long x 5 cm (2 in.) thick. The exhaust flow was through the inside of the tubes. Past experience had shown that even though over 90 percent of the water vapor may have been condensed out, there were some minute liquid water droplets (aerosols) suspended in the condenser exhaust which appeared as ice fog. One solution was to impinge the droplets upon a surface causing them to coalesce and run off in a liquid stream. A coalescer was fabricated out of a 61 cm (24 in.) section of 10 cm (4 in.) stove pipe into which a 7.6 cm (3 in.) plug of expanded fiberglass furnace air filter was placed. By means of a 4.4 cm (1-3/4 in.) flexhose, the condenser exhaust was directed at the fiberglass coalescer in the 10 cm (4 in.) pipe. The setup is show in Figure A-6. The coalescer effectively removed the droplets without freezing shut. Visible ice fog at -28°C (-20°F) was negligible. The cooler-condenser was more than adequate. There were only two problems with this application, one was excessive heat and the other was plugging. The cooler-condenser was mounted next to a thermoplastic parking light which partly melted. The plastic grill was unaffected. During long, high speed trips, the condensate would freeze in the tubes, restricting them, causing enough back pressure to reduce engine performance. Apparently the tubes which were 1.9 cm (3/4 in.) x 0.16 cm (1/16 in.) in cross section were too small to drain before they froze.

SINGLE PASS STAINLESS STEEL COOLER-CONDENSER

During the winter of 1975-76 a new cooler-condenser was built for the 1974 Chevy Nova sedan using 1 cm (1/2 in.) diameter type 409 stainless steel (Figure A-7). Engineering design and tubing were furnished by AERS and a fabrication contract awarded to the University of Alaska, Geophysical Machine Shop. Price for fabrication was $600. The tubing ends were beaded with a parker beading tool and swaged into the headers. The unit was 89 cm (35 in.) long with 25 tubes. Its surface area was 0.85 square meters (9.1 square feet).

It was mounted between the radiator and grill on the front of the vehicle. 2.9 meters (9.5 ft.) of flexible exhaust hose were used to pipe the hot exhaust from the muffler inlet to the cooler-condenser inlet header. A chromel alumel thermocouple was inserted in the inlet header. The outlet was piped to a coalescer mounted under the bumper in front of the driver's side. Later a 4.4 meter (14.5 ft.) length of flexhose was extended to the rear of the vehicle adding 0.85 square meters (6.2 square ft.) surface and a coalescer with an iron constantan thermocouple was inserted in the outlet exhaust flow.

At idle the output temperature averaged 10°C (50°F) with inlet temperatures of 180°C (350°F). At speeds of 64 km/h (40 mi/h) the output temperature averaged 21°C (70°F). Back pressure at idle was 3-10 cm (1-4 in.) of water. At 64 km/h (40 mi/h)it was 76 cm (30 in.) of water.

Each tube had chains inserted inside to increase internal surface area and to turbulate the flow of gases. The cooler-condenser was over-sized in design (flow area) to get the required surface area. Therefore, addition of chains did not affect engine performance or increase back pressure.

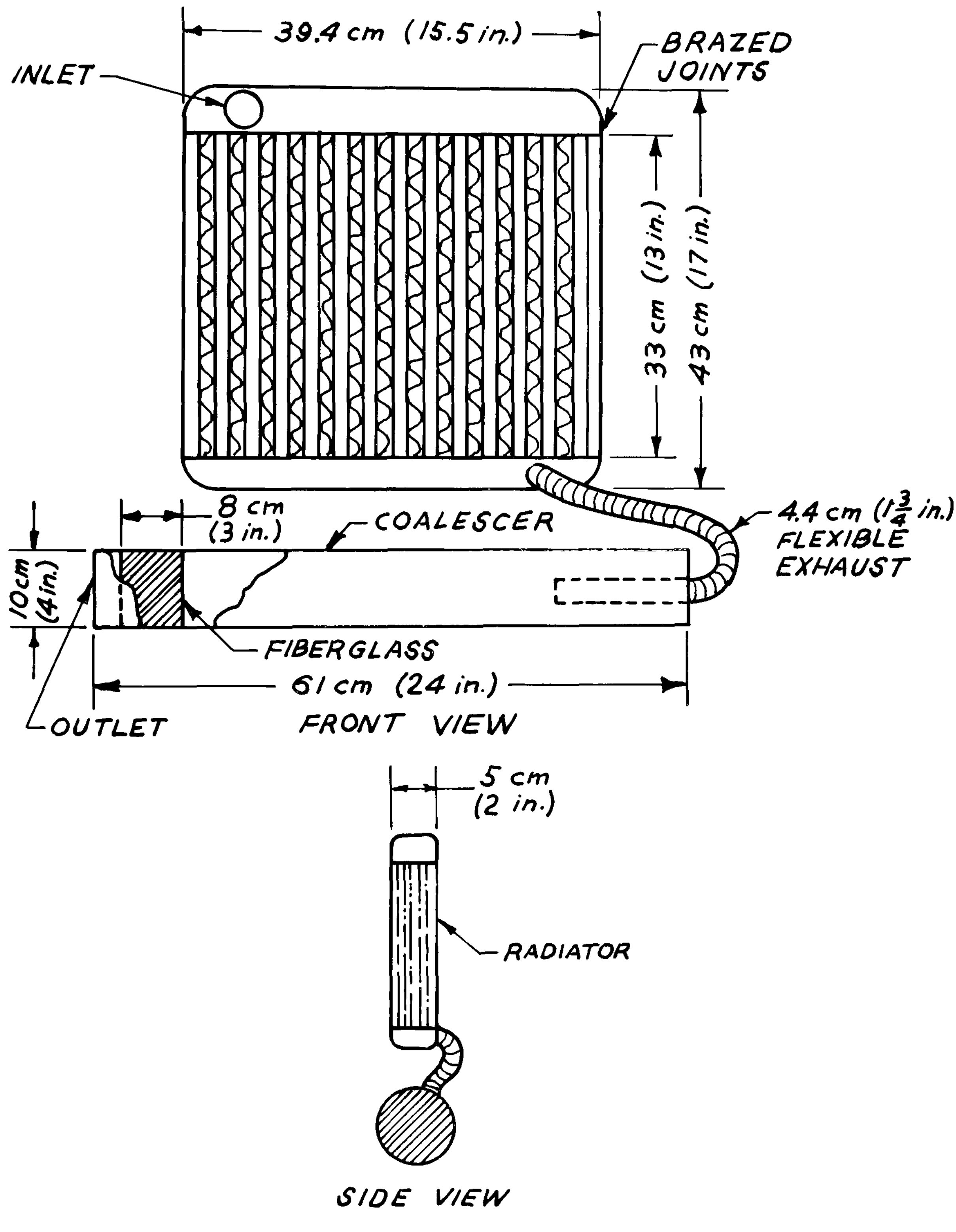

Figure A-6. Brazed radiator cooler-condenser and coalescer on 1974 Chevrolet Nova

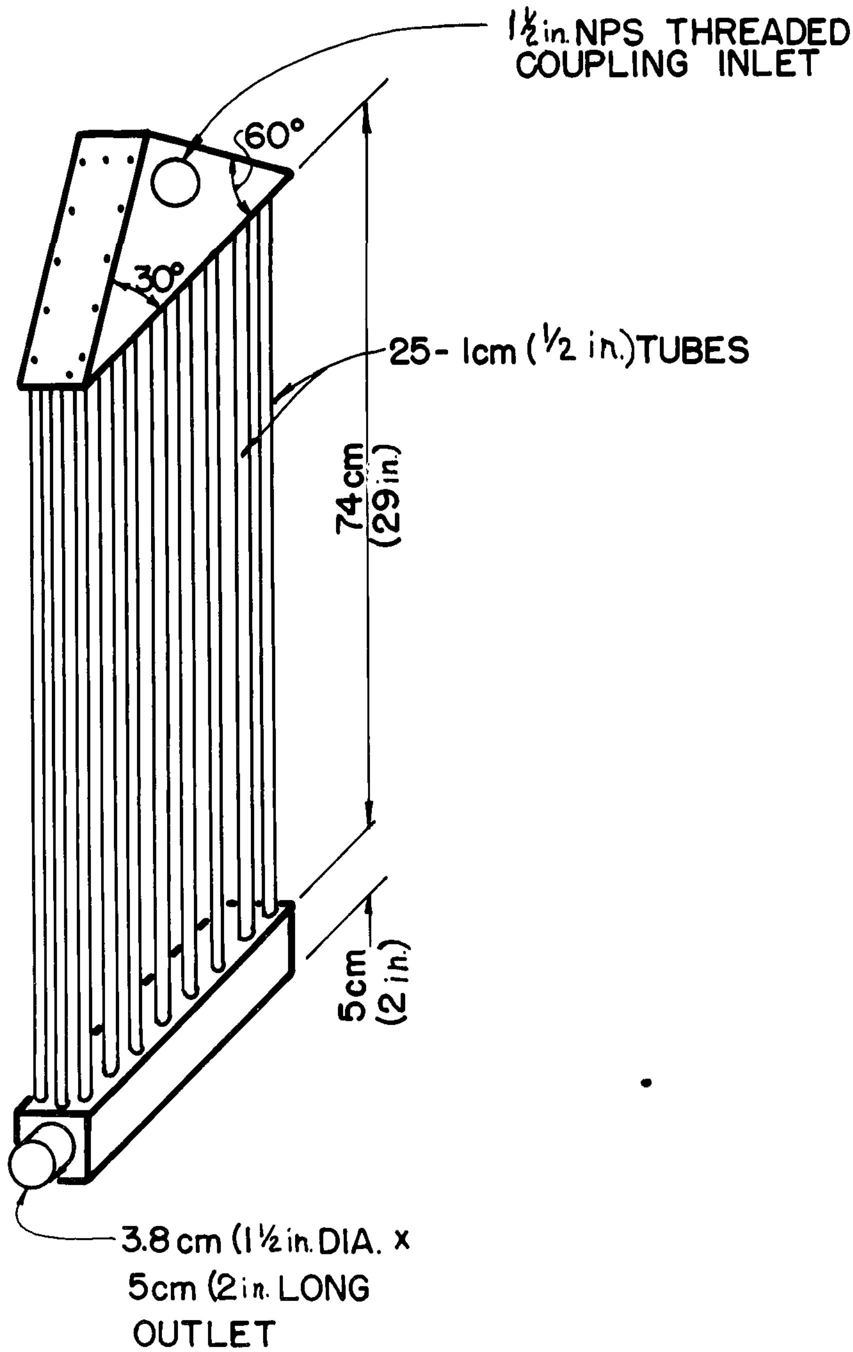

Figure A-7. Single pass stainless steel cooler-condenser on 1974 Chevrolet Nova

MODIFIED FINNED OIL COOLER COOLER-CONDENSER

Demonstrations were progressing satisfactorily with the gasoline combustion engines so it was decided to try a vehicle equipped with a diesel engine. A 1967 Mercedes Benz D200 was volunteered. The exhaust cooler-condenser for this vehicle was center mounted between the grill and the radiator. In this application, the exhaust was piped directly from the manifold to the cooler-condenser. The exchanger used was a Young Radiator Company mobile oil cooler #2 (MOC-2) as shown in Figure A-8. At first, the MOC-2 was not modified, but after measuring excessive pressure drop, the internal tube tubulators were removed. The overall dimensions of the MOC-2 were 41 cm (16 in.) wide x 43 cm (17 in.) long x 3.8 cm (1-1/2 in.) thick.

This method worked satisfactorily in that the visible ice fog diminished within a foot or so of the automobile at temperatures below -18°C (0°F). This allowed total visibility of the vehicle during hazardous winter driving conditions.

One problem encountered was the smell of exhaust fumes inside the cab compartment. Therefore, it is important for the comfort and safety of the passengers that the front mounted cooler-condensers be leak proof and the exhaust outlet extend so it will bypass the seating area. Exhaust leaking out of weep holes should be directed such that it will flow under the passenger compartment since the heater system usually takes in fresh air between the hood and windshield.

FOUR PASS STAINLESS STEEL COOLER-CONDENSER

A new cooler-condenser was designed for the Mercedes during the winter of 1975-76 using the calculated amount of square feet as a guide for maximum efficiency. Stainless steel type 409 was selected for the 1 cm (1/2 in.) tubing and 18 gauge stainless steel sheets were formed into two multi-chamber headers. The direction of flow through each row of tubes switched after entering each chamber. There were four flow passes; see Figure A-9. The last row of tubes had chains inside each tube to increase turbulance and internal surface area.

The material and design were furnished by AERS, and Midway Welding of North Pole did the welding. Cost of fabrication was $700. Installations of connecting piping were performed by the Fairbanks Muffler Shop at a cost of $150.

The condenser was mounted on the front of the vehicle between the radiator and grill. A short exhaust pipe was connected directly to the engine's manifold and secured tightly to prevent any gases from being drawn into the passenger compartment. The drawback of using a short rigid pipe was that it transmitted engine vibration directly to the auto body, bypassing motor mounts. Therefore, a one foot section was replaced with flexhose wrapped with high temperature asbestos tape to prevent gas leaks and to reduce transmission of motor vibration. The tape failed to eliminate fume leakage, but it was satisfactory for testing purposes.

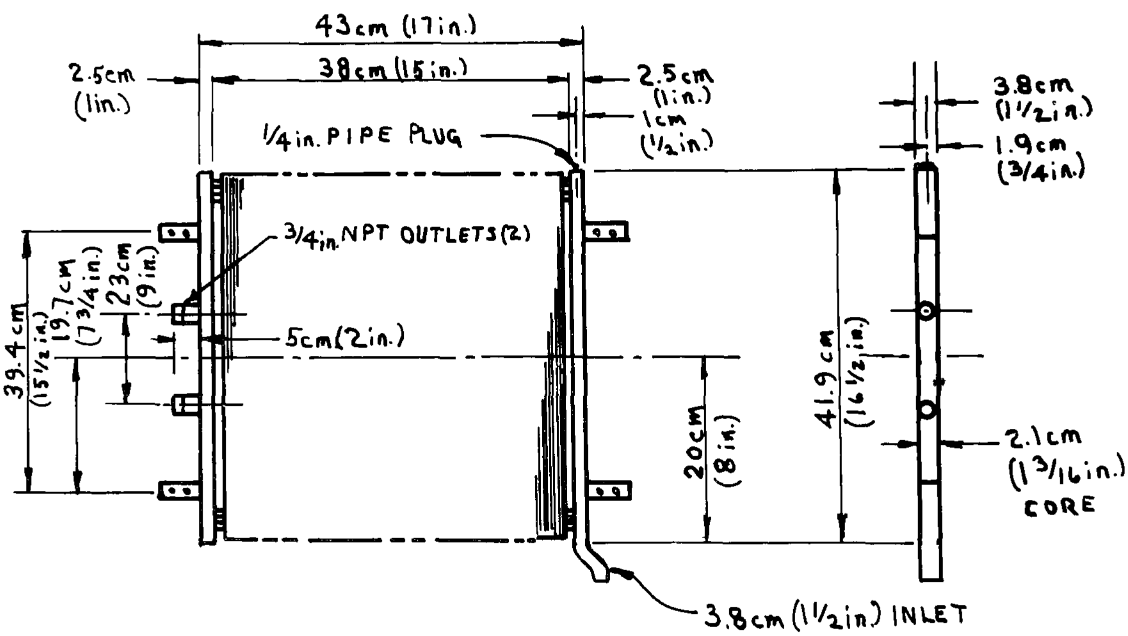

Figure A-8. Modified finned oil cooler (MOC-2) cooler-condenser on 1967 Mercedes Benz.

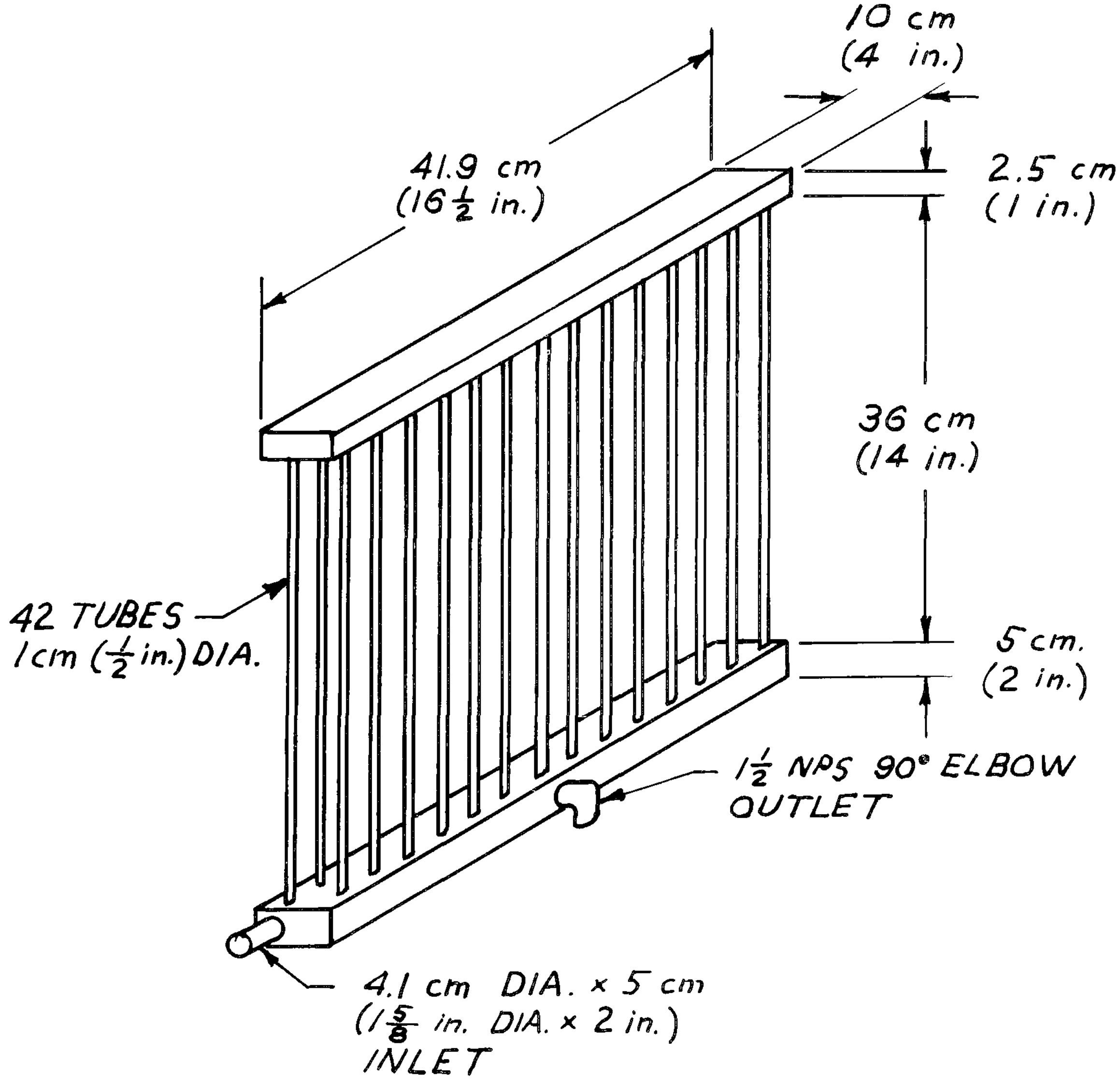

Figure A-9. Four pass stainless steel cooler-condenser on 1967 Mercedes Benz.

A six foot section of plastic suction hose connected the cooler-condenser outlet to the muffler. A plastic hose was used to reduce the amount of condensate freezing to the inside surface and increasing back pressure. Water freezes to a cold metal surface more readily than to plastic. Measurements of outlet temperatures indicated that internal freezing was not to be considered a problem so the plastic hose was removed and replaced with metal flexhose, adding 0.3 square meters to the system, bringing the total surface area to 2.3 square meters (24-1/2 square feet). A coalescer was mounted to the back bumper with an iron constantan thermocouple inserted into the exhaust stream. This coalescer was a 36 cm (14 in.) length of 7.5 cm (3 in.) plastic pipe with a 7.5 cm (3 in.) thick fiberglass coalescing medium. The fiberglass became plugged with soot and ice.

Inlet temperatures ranged between 93°C (200°F) at idle to over 260°C (500°F) at 64 Km/h (40 mi/h). Back pressure measured between 200 and 500 kdy/cm^2 (3 and 7 psig). Ambient air temperatures on different testing days were from -15°C (5°F) to -40°C (40°F).

The cooler-condenser effectively removed approximately 80 percent of the visible ice fog. Under heavy acceleration, a plume of water particles was visible at the tail pipe.

Some power loss was observed while climbing hills, indicating excessive back pressure. Removal of the unit and gathering amounts of condensation which drained out after allowing it to thaw overnight only produced about 200 ml of water. This was not considered enough to account for the power loss. Therefore, excessive back pressure had been built into the cooler-condenser.

Since the device was designed to calculated specifications, further testing was required to determine the cause for the increased back pressure. The problem was narrowed down to the chains which had been added to the last row of tubes. They had not been taken into account during the original calculations and were introduced as a last minute suggestion.

A mock-up was built using a positive displacement rotary blower to provide sufficient air flow to simulate the engine's exhaust. This was attached to a flow meter in series with a piece of 1 cm (1/2 in.) 409 stainless steel tubing cut to the length of those used on the cooler-condenser. A pressure gauge was attached immediately preceding the tube. Tests were run with and without internal chains. At a constant air volume of 10 cmh (6 cfm) the back pressure without a chain was 5 kdy/cm^2 (2 in. water). At the same volume of air with a chain, the back pressure was 120 kdy/cm^2 (48 in. water), an increase of 2300 percent. Because of warm weather, the unit was not replaced on the vehicle for further testing. However, it was assumed that it would have worked just as effectively without internal turbulators and there would not have been any noticeable power loss due to back pressure during heavy load requirements.

FAN TUBE COOLER-CONDENSER

The Arctic Studies Group designed and fabricated a fan cooled shell and tube cooler-condenser for a 1970 Volvo Sedan. It was fabricated from 16 electrical metal tubes (EMT), 1-1/2 in. x 15 in. enclosed in a 12 in. x 15 in. x 8 in. sheet metal box. A 12v DC fan was mounted at one end to draw ambient air through each tube so a continuous flow could be maintained. See Figure A-10. The cooler-condenser was mounted on the rear bumper of a 1970 Volvo 144S. Exhaust gas at temperatures between 93°C (200°F) and 148°C (300°F) flowed into the shell and was baffled around the tubes. The exhaust gas outlet was recirculated through the tubes, mixing with the ambient air being drawn by the fan. This outlet temperature was in the range of 5°C (40°F).

This device prevented any visible ice fog from being released into the atmosphere. The condensed water remained in a liquid state and leaked out of the condenser through the seams, freezing on contact with the road.

Simulated tests were performed at the University's cold cell laboratory where room temperatures of -26°C (-15°F) to -31°C (-25°F) were maintained. A steam generator was used to simulate automotive exhaust. This ingoing imitation exhaust temperature measured between 93°C (200°F) and 148°C (300°F). Test results produced no visible exhaust vapor in the form of ice fog. However, the simulated test data could not duplicate road conditions.

The time involved for assembling required approximately 4 hours in construction time, 1/2 hour for installation, 2 hours for simulated testing, and 2-1/2 days of road testing while mounted on the vehicle.

Material cost can be itemized as:

Sheet metal	\$15.50
Conduit	12.00
Fan & Motor	20.00
Miscellaneous	4.00
	\$51.50

Taking into consideration the corrosion factor, this system had an anticipated life of no more than three seasons.

FINNED COPPER TUBING COOLER-CONDENSER

Early project plans by AE Research, Inc. involved the fabrication of an air cooled cooler-condenser made from 3.2 cm (1 1/4 in.) aluminum tubing surrounded by 10 cm (4 in.) square fins spaced 2.5 cm (1 in.) apart over a 1.3 m (4 ft.) length (9). A twisted aluminum strap was inserted to create turbulance. It was mounted on the rear of a 1972 Toyota with a 1400 cc engine. This was discarded when access to the Toyota ended.

A 1974 Datsun with a 1300 cc engine was used as a replacement in the project. Since the engine displacement was smaller, a second cooler-condenser was designed. A manifold style of construction was incorporated using three

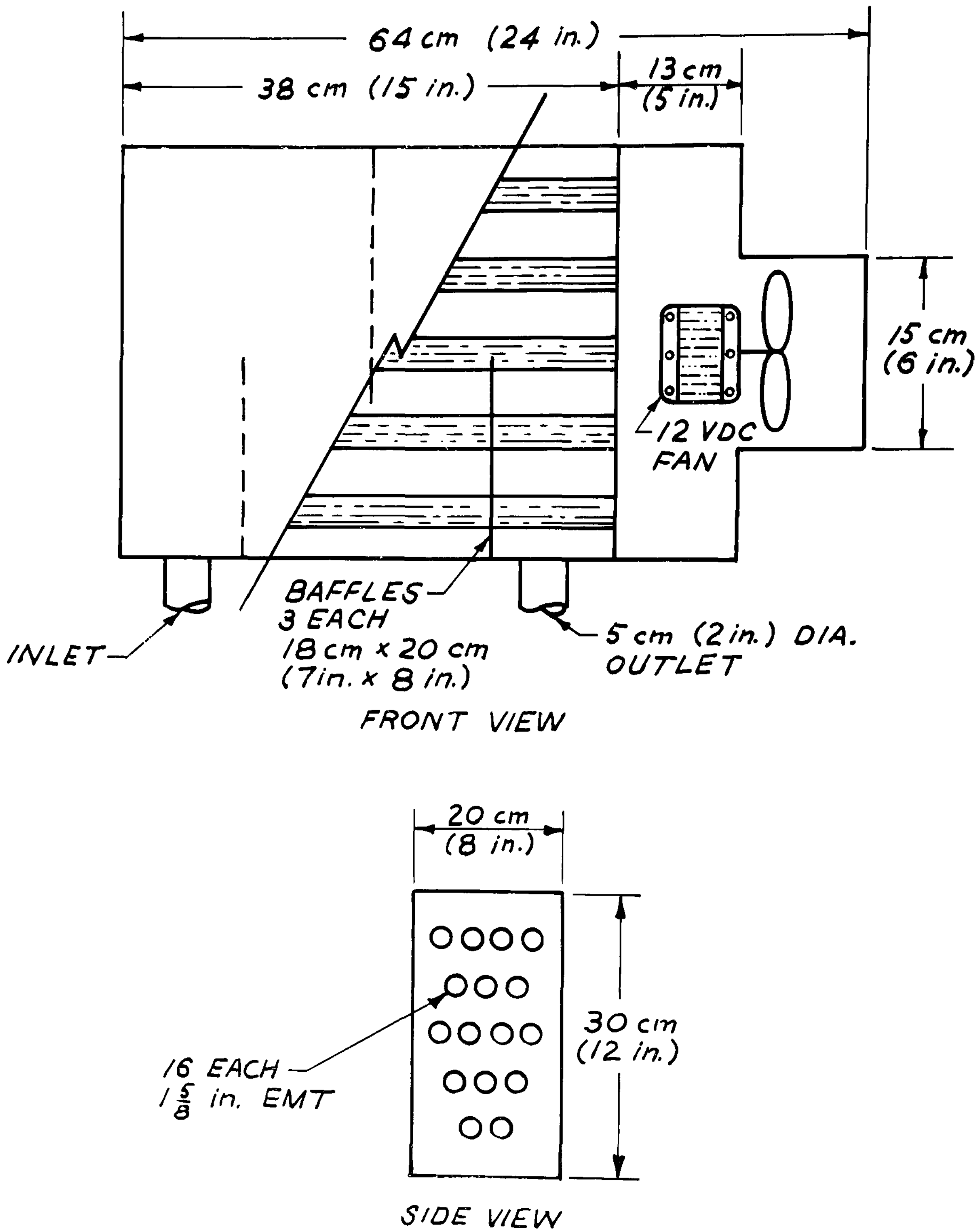

Figure A-10. Fan tube cooler-condenser on 1970 Volvo sedan.

parallel standard aluminum finned-copper tubing baseboard heating pipes 1 m (3 ft.) in length. No internal turbulators were used as the manifold was expected to produce internal turbulance (Figure A-11). The condenser was mounted on the rear of the vehicle at a slight angle to drain condensate from the condenser. Samples of condensate were collected and frozen for analyses. Tests for pH, conductivity, and sulfates were run. pH remained consistent between 3 and 4 indicicating a potentially high corrosive liquid. Conductivity ranged between 500 and 800 μmhos. Sulfates were between 14 and 30 mg/l.

Material cost for the fin tubing and miscellaneous items for construction of the second unit came to about $55. No cost figures were given for the first unit constructed. Installation time for mounting and testing were not included in the report.

Heat loss calculations were done on theoretical assumptions of fuel economy in conjunction with measured temperature differences. Total gas flow rate was measured using an Alnor velometer and exhaust gas density relationships were provided by supplements from the AERS.

At engine idle the percent water vapor condensed approached 100 percent, but not all visible vapor had been dispersed. The remaining aerosol fog was of such a minimal amount that the condenser could be considered a success. However, at higher speeds requiring more fuel consumption, the amount of visible ice fog increased. To function more effectively, a more sophisticated design is required which would pass more air over the fins to take away the heat.

LIQUID COOLED COOLER-CONDENSER

Working with the University of Alaska Mechanical Engineering Department, H & S Research designed a liquid cooled cooler-condenser rather than an air cooled type (10). Their version consisted of a water-antifreeze system mounted on the front of a 1968 Jeep Wagoneer. The coolant was connected in series with the vehicle's cooling system. The exhaust gases were routed into a shell encasing a radiator; Figure A-12. The exhaust temperatures going into the condenser ranged between 93°C (200°F) and 148°C (300°F) while the incoming coolant was about 26°C (80°F). The outgoing coolant and exhast gases measured approximately 37°C (100°F). The outgoing gases were then directed at the vehicle's radiator which reevaporated all water particulates which were not removed in the condenser. This last step was undesirable because of the hazards of drawing poisonous exhaust gases into the vehicle's cab. Also the reevaporated water would form ice fog once it cooled down. Large icicle formations accumulated at the lower seams where the condensed water weeped out into the cold atmosphere. These icicles broke off and fell onto the roadside.

Since safety was a prerequisite along with eliminating visible exhaust, the unit was relocated to the rear of the vehicle. From this location it was no longer practical to use the Jeep's radiator coolant.

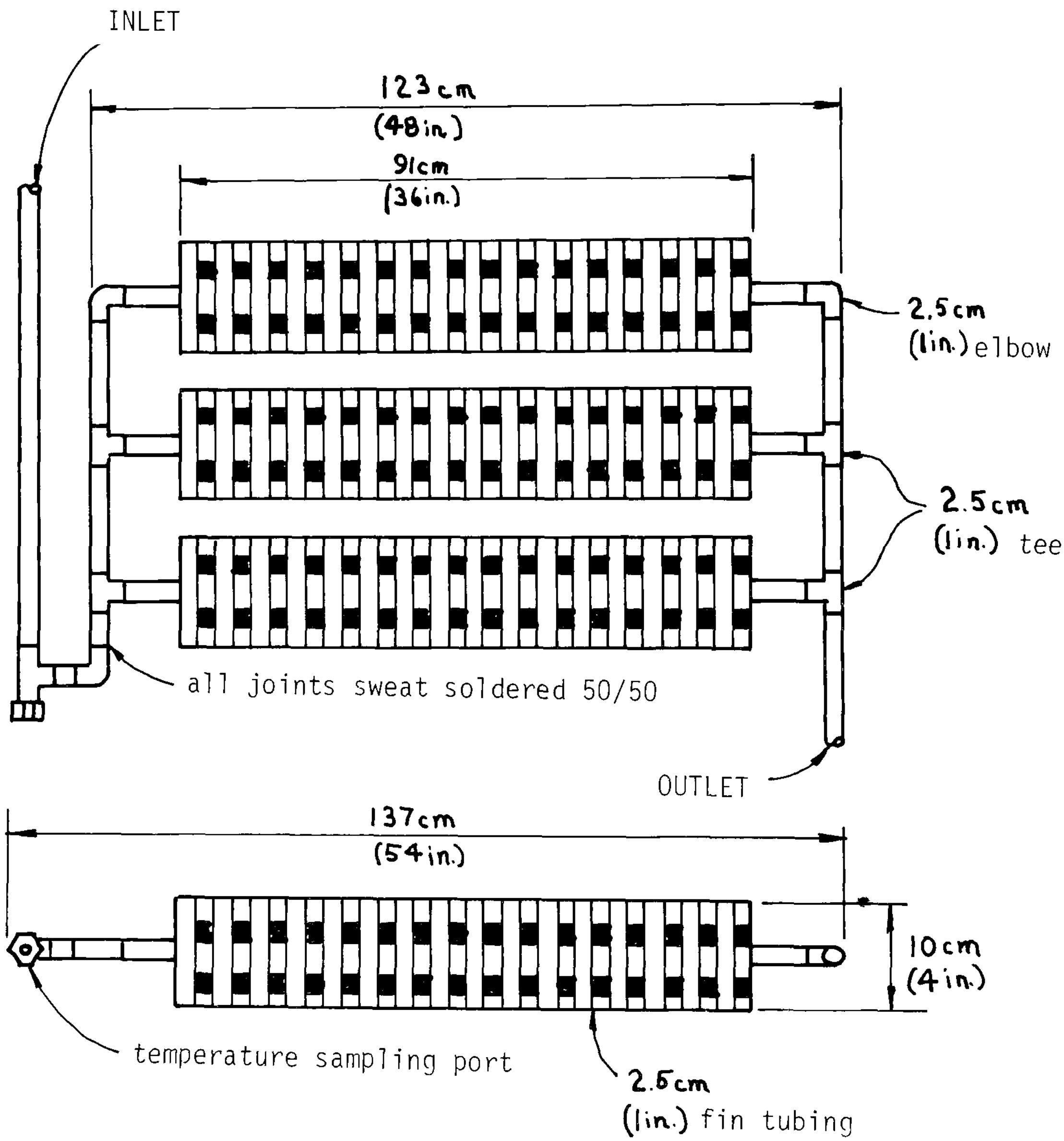

Figure A-11. Finned copper tubing cooler-condenser on 1974 Datsun sedan.

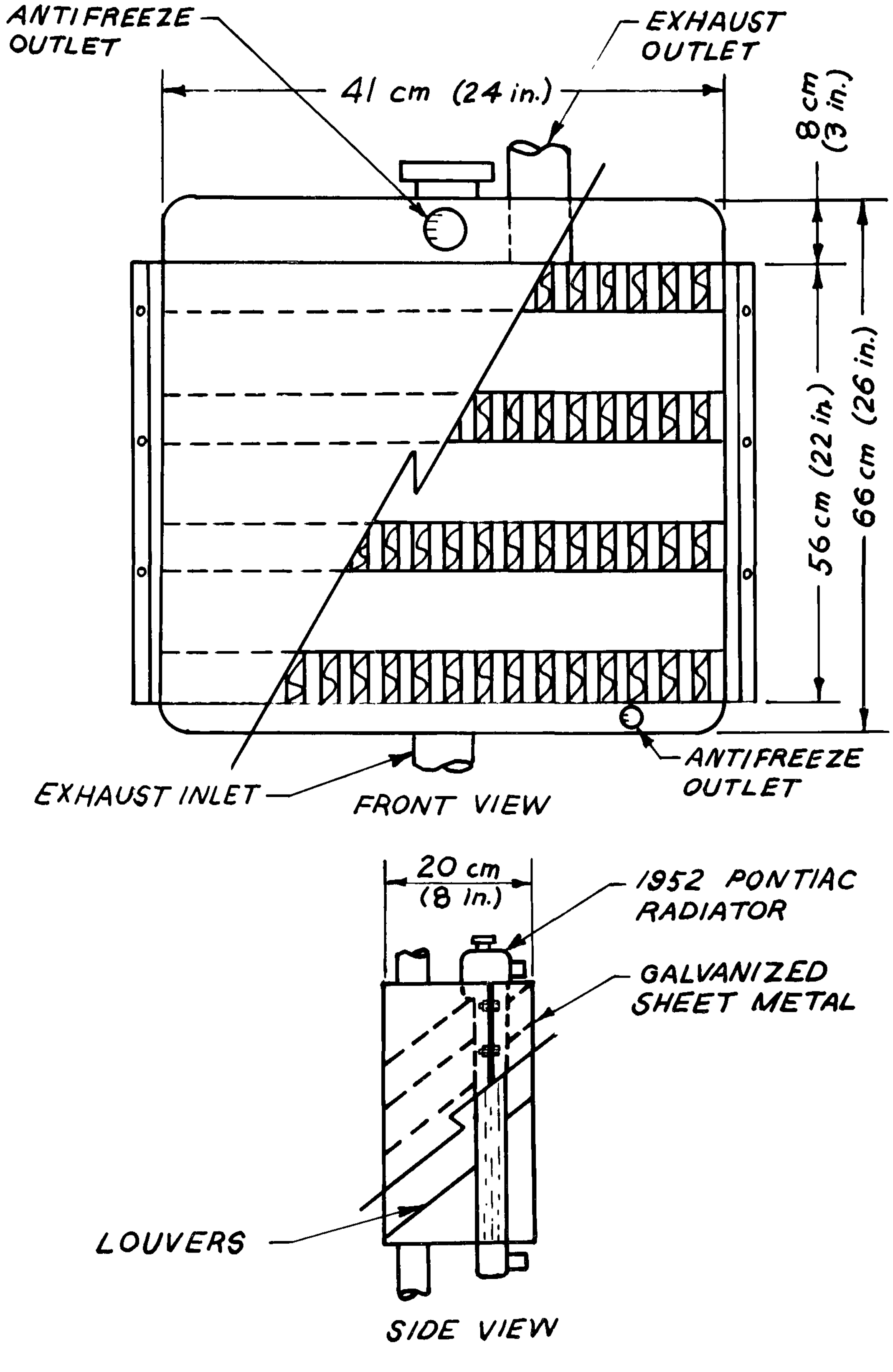

Figure A-12. Liquid cooled cooler-condenser on 1968 Jeep Wagoneer.

Another configuration was developed using a 15 gpm 12v DC pump and a second radiator. Due to warm weather this system was tested at the University's cold cell laboratory with room temperatures of -31°C (-24°F) to -35°C (-31°F). A steam generator was used to simulate automobile exhaust. The ingoing pseudo-exhaust temperature was 121°C (250°F). The outgoing exhaust measured 2°C (35°F) and coolant temperature was 5°C (40°F). There was no visible exhaust vapor with this mock setup.

Material costs for the first device, including antifreeze, totaled $169. Additonal cost for the second device was $135. Total overall material cost for this project was approximately $300.

The flexible exhaust hose was considered the weakest part of the system, due to its susceptibility to corrosion and attack by sulfurous acids. However, the sheet metal shell is equally attacked by acidic condensate. Its anticipated life was no more than two seasons.

FINNED PIPE COOLER-CONDENSER

Scarborough and Associates designed and fabricated a finned pipe cooler-condenser using a 1.3 m (4 ft.) length of 5 cm (2 in.) aluminum pipe with forty 15 cm (6 in.) square aluminum fins around it, spaced 2.5 cm (1 in.) apart (11). A spiral strip of steel was inserted to act as a gas turbulator (Figure A-13). It was mounted after the muffler on a 1968 Chevy Carryall (4x4).

Although simplicity of construction was a major advantage, this condenser appeared to be undersized since it failed to remove any substantial amounts of visible exhaust vapor either at idle or higher speeds.

The ingoing exhaust temperature was measured at 96°C (205°F). The outgoing exhaust temperature was 39°C (102°F) when the ambient air was -20°C (-5°F). This value of exhaust gas temperature was too high to condense most of the water vapor.

Construction and installation cost amounted to approximately $250. Since the unit was custom made and parts were acquired locally, it is speculated that commercially built cooler-condensers of this type could be produced for approximately $80.

The condenser was used in service for a period of 30 days; approximately 1300 miles were accumulated. It was removed and inspected for corrosion and any buildup of deposits. No pitting or other acidic damage was evident. The interior wall and turbular strip had a uniform soot deposit. The aluminum pipe had an anticipated service life of about three years. Recommendations were made for construction using light-weight stainless steel.

LOUVERED SHELL COOLER-CONDENSER

A cooler-condenser fabricated out of 1.2 cm (1/2 in.) EMT (12) was designed by Simplex-Standard to meet the requirement for a 1968 IHC Scout

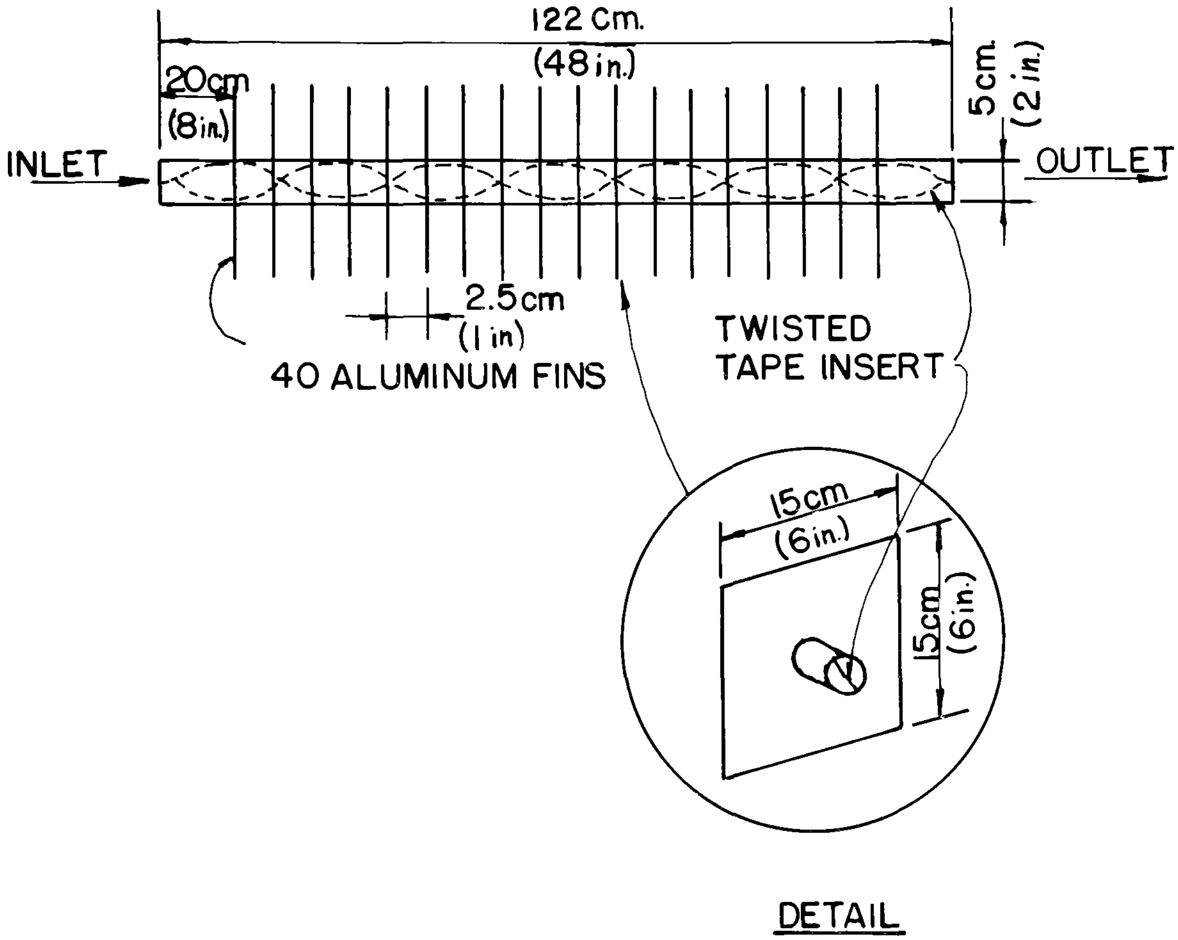

Figure A-13. Finned pipe cooler-condenser on
1968 Chevrolet Carryall (4x4).

equipped with a V8 266 CID engine. For simple construction there was no welding or machining of parts (Figure A-14). The condenser was designed so that the outlet header could easily be removed to allow for cleaning and deposit removal inside the tubes. Mounting was easily accomplished by bolting it onto the front bumper and running a 5 m (15 ft.) piece of flex exhaust hose from the tail pipe to the inlet header.

Air flow across the tubes was controlled from inside the cab by a mechanical cable attached to six adjustable baffles in front of the tubes. Freezing of the two bottom tubes became a problem which was solved by blocking the air flow across them. Materials for the condenser cost approximately $85. Roughly 40 hours were required to fabricate and install the condenser. The unit was tested for five weeks in daily routine traveling. Ambient temperatures ranged from -7°C (20°F) to -37°C (-35°F) and at speeds from zero to 72 km/h (45 mi/h). The average temperature drop across the condenser was approximately 10°C (50°F). Inlet temperatures ranged from 54°C (130°F) under heavy load to 36°C (98°F) at idle. Outlet temperatures were from 28°C (84°F) to 6°C (42°F), respectively. Low inlet temperatures were explained in part by the heat loss through the 5 m (15 ft.) of exhaust flexhose.

An inspection of the tubes after five week's use revealed a small accumulation of scale and powder residue inside the tubes and headers. From this, an anticipated life of three to four seasons could be assumed by using the most inexpensive materials. A considerably longer seasonal life could be expected by using corrosion resistant alloys.

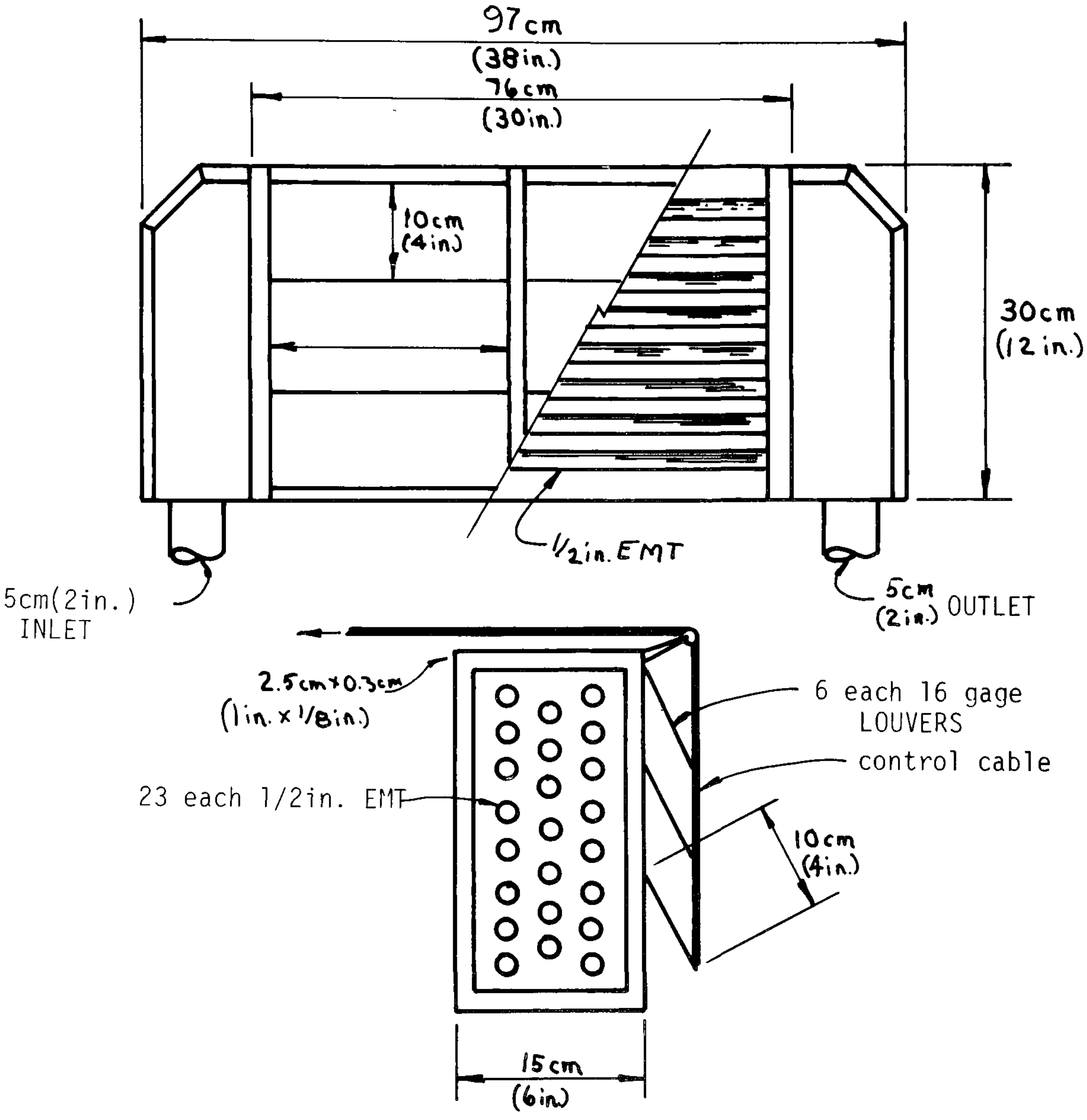

Figure A-14. Louvered shell cooler-condenser on 1968 International Scout

APPENDIX B

CALCULATION OF CONDENSATION CURVE AND HEAT EXCHANGER DUTY

Gasoline is a mixture of many hydrocarbons: it contains butene C_4H_8 at the lighter end, and decane $C_{10}H_{22}$ at the higher boiling, heavier end. The mass ratio of carbon to hydrogen C:H in commercial gasolines varies from 6:1 to 6.8:1.

For this example a gasoline with a C:H of 6.5:1 will be used, its empirical chemical formula is C_7H_{13}.

A complete combustion chemically balanced equation with air is:

	Fuel		Air			Exhaust				
Form:	C_7H_{13}	+	$10.25\ O_2$	$+\ 38.56\ N_2$	$\rightarrow$	$7.00\ CO_2$	$+\ 6.50\ H_2O$	$+\ 38.56\ N_2$	lbs.
lbs:	97	+	328	+ 1080	=	308	+ 117	+ 1080	

Since there are no carbon monoxide or hydrocarbons in the exhaust, the equation assumes complete combustion. Well tuned engines approach complete combustion; therefore, their exhaust water concentration approximates the above equation. The exhaust water vapor [$H_2O_{(v)}$] is the ice fog.

The above equation is based on an air to fuel weight ratio (A/F) of (328 + 1080)/ 97 = 14.5:1, which is called STOICHIOMETRIC.

Most gasoline fueled automobiles operate at A/F between 12:1 and 16:1. For purposes of sizing an exhaust gas cooler-condenser the heat contents and water vapor condensation will be based upon the combustion of one pound of gasoline. At the stoichiometric ratio it will yield 15.5 pounds of exhaust gas of the following composition: 3.18 lb. CO_2, 1.21 lb. H_2O and 11.14 lb. N_2.

The reference temperature for heat content of the noncondensables - CO_2 and N_2 is 60°F. For H_2O, 32°F is used.

Table B-1 Heat Content at 300°F, 149°C

Component	BTU/lb°F		lb		ΔT°F		BTU
CO_2	0.215	x	3.18	x	240	=	164
N_2	0.249	x	11.14	x	240	=	667
$H_2O_{(v)}$	4.46	x	1.21	x	268	=	1447
			15.5				2278

As the exhaust gas is cooled, the water vapor starts to condense at the dew point temperature. The dew point is reached when the vapor pressure (VP) of water equals its partial pressure (mole fraction x total pressure). Assume the pressure drop in the condenser to be 0.4 psi. Since at 500 feet above sea level the atmospheric pressure is 14.5 psia then total presure (P) half way through the condenser is 14.7 psia.

Dew point temperature (T_{dp}) is temperature when:

$$VP = P \times \frac{H_2O}{CO_2 + H_2O + N_2}$$

$$VP = (14.7) \times \frac{6.50}{7.00 + 6.50 + 38.56}$$
$$= (14.7)\ (0.125)$$
$$= 1.84 \text{ psia}$$

From steam table: the temperature at which VP = 1.84 psia is 123°F; therefore, T_{dp} = 123°F.

Table B-2 Mole weight and heat content @ 123°F, 51°C

Component	lb./mole wt.		moles	BTU/lb°F		lb		ΔT°F		BTU
CO_2	$\frac{3.18}{44}$	=	0.072	0.2047	x	3.18	x	63	=	41
N_2	$\frac{11.14}{28}$	=	0.398	0.2488	x	11.14	x	63	=	174
$H_2O_{(v)}$	1.21	=	0.067	12.25	x	1.21	x	91	=	1349
Totals @ 123°F		=	0.537			15.5				1564

Mole weight: $\frac{15.5}{0.537} = 28.8$ lb/lb mole

$$\text{Moles } H_2O_{(v)} = \frac{(CO_2 + N_2)\ VP}{(P - VP)}$$

At 60°F lb $H_2O_{(v)}$ = 18 x moles $H_2O_{(v)}$ $= \frac{(0.470)\ (0.256)\ (18)}{(14.7\ \ 0.256)} = 0.149$

$H_2O_{(L)}$ is liquid water (condensate)

$H_2O_{(L)} = H_2O$ (total) $\ \ H_2O_{(v)}$

therefore $H_2O_{(L)} = 1.21 - 0.149 = 1.06$

Amount condensed is $\frac{1.06}{1.21} \times 100 = 88\%$

Table B-3 Heat content @ 60°F, 16°C

Component	BTU/lb°F		lb		ΔT°F		BTU
CO_2	0.20	x	3.18	x	Ø	=	Ø
N_2	0.24	x	11.14	x	Ø	=	Ø
$H_2O_{(v)}$	38.9	x	0.149	x	28	=	162
$H_2O_{(L)}$	1.01	x	1.06	x	28	=	30
							192

For brevity many calculation steps have been omitted. The results of the above computations have been plotted on Figures 1 and 2 (pages 9 and 20). Similar calculations were performed for exhaust from diesel and propane fueled engines in order to draw their respective condensation curves.

The diesel is the only piston engine that normally operates with excess air. This excess air results in a lower exhaust moisture concentration; hence a larger fraction will remain in the vapor phase at any given temperature when compared to exhausts with no excess air.

APPENDIX C

HEAT EXCHANGER DESIGN TECHNIQUES

An automotive ice fog cooler-condenser is simply a heat exchanger with exhaust gas on the hot side and ambient air or other coolant on the cold side.

Heat exchangers are specified (sized) by the amount of surface area (square feet) required to adequately transfer the heat. For forced convection heat transfer, the surface, A, is related to: the amount of heat to be transferred, Q; the temperature difference, ΔT; and the overall heat transfer coefficient, U. This realtionship can be expressed thusly:

$$\text{Transfer surface - ft.}^2\text{, } A = \frac{Q}{U\Delta T_L}$$

Where: the units of Q (the exchanger duty) are BTU/hr.,
the units of ΔT_L, (the log mean temperature difference-LMTD) are °F
the units of U are BTU/hr.ft.^{2}°F.

Since U is the overall transfer coeficient, its reciprocal 1/U is the overall thermal resistance.

To size the exchanger (A) one needs to know the duty, the heat transfer coefficient, and the temperatures. First the exchanger duty Q will be calculated. It depends upon the exhaust gas flow rate and temperature. A vehicle's exhaust flow is directly related to its fuel economy. For example, say at 40 miles per hour a vehicle gets 20 miles per gallon and its exhaust temperature is 500°F. At -25°F the density of gasoline is 6.4 pounds per gallon; the gasoline consumption rate is therefore:

$$6.4\ \frac{\text{lb}}{\text{gal}} \times \frac{1\ \text{gal}}{20\ \text{mi}} \times 40\ \frac{\text{mi}}{\text{hr}} = 12.8\ \frac{\text{lb gasoline}}{\text{hr}}$$

From Figure 1 the heat content of the exhaust is 3280 $\frac{\text{BTU}}{\text{lb gasoline}}$ at 500°F and 80 $\frac{\text{BTU}}{\text{lb gasoline}}$ at 40°F. Therefore the duty, Q is 12.8 lb/hr. x 3200 BTU/lb = 41,000 BTU/hr. Note that at 40°F approximately 94 percent of the exhaust water vapor has been condesnsed out. The calculations for making Figure 1 are in Appendix B.

Next, consider heat transfer through a metal tube wall. The overall heat transfer coefficient U needs to be calculated. Its reciprocal, the overall thermal resistance 1/U, is estimated from empirical data. 1/U is the sum of several series resistances, expressed in the following formula:

$$\frac{1}{U} = \frac{A_o}{A_i\ h_i} + \frac{xA_o}{K\ A_{ave}} + \frac{1}{h_f} + \frac{1}{h_o}$$

Where: A_i = inside surface area for heat transfer.
A_o = outside surface area for heat transfer.
A_{ave} = $(A_o + A_i)/2$
h_i = inside film coefficient for heat transfer.
h_o = outside film coefficient for heat transfer.
h_f = fouling film coefficient for heat transfer.
k = thermal conductivity of the tube wall.
x = thickness of the tube wall.

The reciprocal 1/h etc. are the respective thermal film resistances. The term $xA_o/k\ A_{ave}$, which is the thermal resistence of the metal wall is negligible compared to the other resistances and therefore can be ignored.

For the thin wall tubes A_o/A_i will be about 1.0. The film conductance depends upon film composition, surface roughness, temperature, and fluid velocity. The individual film coefficients are estimated by procedures detailed in the Engineering Manual, Reference 13 (pages 2-65,70., tables 2-14, 15,. case 11 and 20). Assuming an average exhaust temperature of 200°F, the inside film base factor is 4.1. A gas velocity of 58 ft/sec yields a correction factor of 2.3 for 1/2 in. diameter tubes. Therefore, $h_i = 4.1 \times 2.3 = 9.4$.

Assuming ambient air temperature of 15°F, the outside film base factor is 7.7. An air velocity of 40 mph yields a correction factor of 3 for 1/2 in. diameter tubes. Therefore, $h_o = 7.7 \times 3 = 23$.

Lead and soot deposits will foul the inside surface, therefore a fouling film coefficient (h_f) of 500 is used. Substituting the coefficients into the above formula:

$$\begin{aligned} 1/U &= 1/9.4 + 1/500 + 1/23 \\ &= 0.1064 + 0.002 + 0.0435 \\ &= 0.1519 \end{aligned}$$

Therefore U = 6.6 BTU/hr-ft^2-°F

Heat transfer by radiation is neglected.

Last, it is necessary to calculate the log mean temperature difference (LMTD). Assume the condenser is to work at ambients of -15°F or less at 40 mph. So much air will be flowing across the condenser that the air temperature will increase less than 5°F.

$$\text{LMTD} = \frac{\text{Greatest temp. diff. (GTD)} \quad \text{least temp. diff. (LTD)}}{\log_e \text{GTD/LTD}}$$

$$\text{Therefore, LMTD} = \frac{[500 - (-15)] \quad [40 - (-15)]}{\log_e (500 \;+\; 15/40 + 15)} = \frac{460}{2.24} = \quad 206°\text{F}$$

Now the surface area required for heat transfer can be calculated:

$$A = \frac{Q}{U\ (\text{LMTD})} = \frac{41{,}000}{6.7\ (206)} = 30\ \text{ft}^2$$

In evaluating the overall coefficient it can be seen that the largest resistance is the inside film, $1/h_i$. If the inside surface and velocity were increased by adding a metal twisted tape insert in each tube or if other turbulators were added, then the effective inside heat transfer coefficient (h_i) based on the outside surface could be increased to about 18 BTU/hr-ft^2-°F.

The resulting overall coefficient and required transfer area would then be:

$$U = \frac{1}{1/18 + 1/500 + 1/23} = 9.9$$

$$A = \frac{41{,}000}{9.9\ (206)} = 20\ \text{ft}^2$$

The back pressure drop also needs to be calculated. The condenser cannot practically be removed for higher speed highway driving. The design data for back pressure at 55 mph are 12 mi/gal and 500°F exhaust:

Exhaust flow = 6.4 lb gasoline/gal x 1 gal/12 mi x 55 mi/hr x 15.5 lb exhaust/lb gasoline = 455 lb/hr = 0.13 lb/sec.

Using the ideal gas law for density (reciprocal specific volume, $1/v_s$):

$$1/V_s = \frac{\text{Pressure (atm) x mole wt. } (\frac{\text{lb}}{\text{lb mole}})}{\text{Gas constant (0.73) x temp. (°F + 460)}}$$

$$1/V_s = \frac{1.0\ (28.9)}{0.73\ (960)} = 0.041\ \text{lb/ft}^3$$

The condenser flow area will be the same as the most common tail pipe, generally around two inches in diameter. Its flow area is 0.022 ft^2. The exhaust gas velocity is:

$$0.13\ \text{lb/sec x } 1/0.041\ \frac{\text{ft}^3}{\text{lb}} \times \frac{1}{0.022\ \text{ft}^2} = 144\ \text{ft/sec.}$$

Assume the cooler-condenser tubes will be designed so that the 144 ft/sec velocity will not be exceeded. It will have two sharp (90°) orifice bends, giving a pressure drop of 1.9 velocity heads each. Assume those plus other losses to total 5 velocity heads (K = 5).

g = gravitational constant, 32 ft/sec^2.

$$\text{Pressure drop} = K\,\frac{v^2}{2g} = 5\frac{(144)^2}{64} = 1600 \text{ ft. of gas}$$

$$= 1600 \text{ ft} \times 0.041 \text{ lb/ft}^3 = 67 \text{ lb/ft}^2 = 0.46 \text{ psi}$$

$$= 13 \text{ inches of } H_2O$$

This addition to the normal exhaust system, whould be tolerated since normal back pressures may run 30 to 50 inches H_2O at 50 to 60 mph. Actual total back pressure will be less when the cooler-condenser is used in lieu of the muffler.

APPENDIX D

ESTIMATION OF ROAD ICING FROM ICE FOG CONTROLS ON AUTOMOBILES

The condensate from the cooler-condensers will most probably drop to the road surface and form ice. In some cases the condensate droplets may freeze before striking the road and roll off to the side out of the traffic lanes.

The amount of ice (condensate) formed depends upon four variables:

fuel economy - miles per gallon
cooler-condenser efficiency - percent water condensed
traffic density - vehicles per day
traveled road width - feet

The following ice accumulation calculations are for two cases, one for a straight section of road where the average fuel economy is 16 miles per gallon and the other a heavily traveled intersection where the fuel economy is 8 miles per gallon.

CASE 1

A two lane road with a traveled width of 40 feet, a traffic density of 12,000 vehicles per day, all equipped with cooler-condensers that condense out 85 percent of the exhaust water vapor. The average condensate yield is then:

$$\frac{1\ \text{gal gas}}{16\ \text{veh mile}} \times \frac{1\ \text{mile}}{5280\ \text{ft}} \times \frac{6.4\ \text{lb gas}}{\text{gas gas}} \times \frac{1.21\ \text{lb}\ H_2O}{\text{lb gas}} \times \frac{1\ \text{ft}^3\ H_2O}{62.4\ \text{lb}H_2O} \times \frac{1}{40\ \text{ft}} \times$$

$$12{,}000\ \frac{\text{veh}}{\text{day}} \times 12\ \frac{\text{inches}}{\text{ft}} \times 30\ \frac{\text{day}}{\text{month}} \times 0.85\ \text{efficiency} \times 4\ \text{months} = 0.54\ \text{in.}$$

CASE 2

A heavily traveled intersection such as the College Road, University Avenue intersection which is loaded at 25,000 vehicles per day. It had three four-lane roads, each 60 feet wide, and one two-lane road 40 feet wide. Total width is therefore 220 feet. Because of the lower average speed at an intersection, the cooler-condenser efficiency will be 90 percent.

$$\frac{1\ \text{gal gas}}{8\ \text{veh mile}} \times \frac{1\ \text{mile}}{5280\ \text{ft}} \times \frac{6.4\ \text{lb gas}}{\text{gal gas}} \times 1.21\ \frac{\text{lbH}_2\text{0}}{\text{lbgas}} \times \frac{1\ \text{ft}^3\ \text{H}_2\text{0}}{62.4\ \text{lbH}_2\text{0}} \times \frac{1}{220\ \text{ft}} \times$$

$$25{,}000\ \frac{\text{veh}}{\text{day}} \times 12\ \frac{\text{inch}}{\text{ft}} \times 30\ \frac{\text{day}}{\text{month}} \times 0.9\ \text{efficiency} \times 4\ \text{month} = 0.42\ \text{inch}$$

For comparison, consider what Mother Nature puts on the ground during the four winter months which have average temperatures of less than 5°F. The precipitation is, in inches of water equivalent:

November:	0.69
December:	0.59
January:	0.90
February:	0.49
	2.67

For the two cases the condensate on the road in relation to precipitation will be:

$$\frac{\text{Condensate}}{\text{(Nov. - Feb.) Precipitation}} = \quad \text{CASE 1: } \frac{0.54 \times 100}{2.67} = 20\% \qquad \text{CASE 2: } \frac{0.42 \times 100}{2.67} = 16\%$$

Therefore, if the cooler-condensers were in common use the additional on-the-road precipitation over the natural accumulation would be approximately 16 to 20 percent.

It is a conservative assumption that the condensate would spread out evenly over the total width of the road. If, on the other hand, the accumulated ice formed ridges between the tire tracks it could easily be removed by snow plow, particularly in the left-hand turn lane where the greatest amount of ice accumulates.

EXAMPLE ILLUSTRATING USE:

Air that is just forming ice fog, saturated at -32°C is heated to -18°C. How much moisture can it accept before reforming ice fog (resaturated)?

SOLUTION:

At -32°C the saturated humidity ratio is 0.0002 gmH_2O/gm dry air.
At -18°C the saturated humidity ratio is 0.0008 gmH_2O/gm dry air.
Therefore 1.0002KG of saturated air warmed from -32°C to -18°C can accept 0.0006KG of addition moisture before becoming supersaturated (forming ice fog).

Chart from Figure 3, McFadden, T., Ice Fog Suppression - A Review of Techniques, The Northern Engineer 7(4):29 University of Alaska, Fairbanks, Alaska.

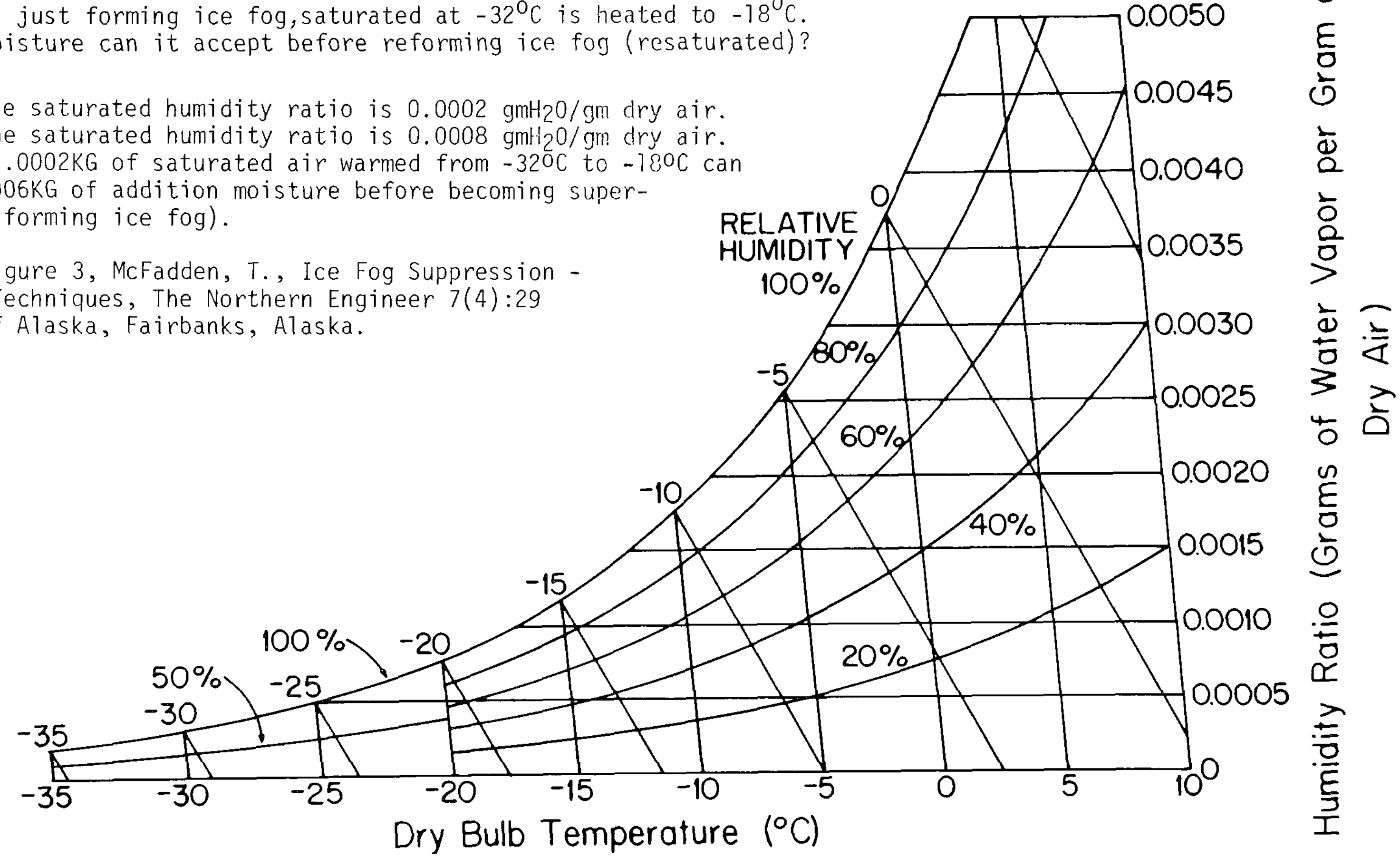

Figure E-1. Low Temperature Psychrometric Chart (Metric Units)

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

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15. SUPPLEMENTARY NOTES

16. ABSTRACT

Automotive generated ice fog is a form of air pollution that results when exhaust water vapor freezes into minute particles which form a dense fog.

The major control technique evaluated was cooling the exhaust gases to well below the dew point, thus condensing water vapor into a liquid stream before final discharge.

During the winters of 1974-75 and 1975-76 the Arctic Environmental Research Station evaluated 12 cooler-condensers on nine inservice vehicles. It was found that ice film formation decreased heat transfer efficiency. An ice fog mass emission reduction up to 80 percent was attained with cooler-condensers on motor vehicles. However, the increase in visibility over roads was not proportional because of the many other ice fog sources. The overall impact of automotive ice fog control would be a visibility increase of at least 70 percent in areas where motor vehicles create 50 percent or more of the ice fog.

Control of automobile-generated ice fog would also mean cleaner air, but perhaps more ice on the road. Cleaner air would result because sulfur oxides and lead compounds would be absorbed in the condensate. This condensate, if allowed to drip from the cooler-condensers, would freeze onto the road and require a more intense snow removal effort.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
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